SAE journal

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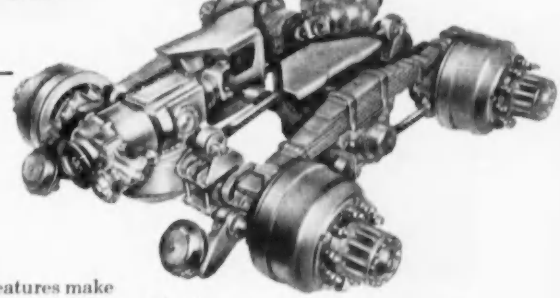

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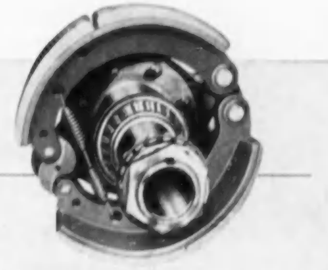

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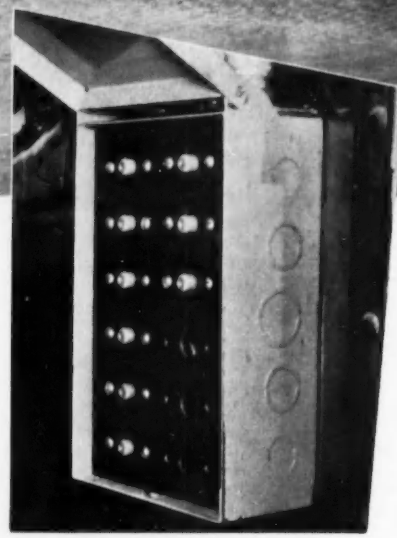

Panel of 9 KLIXON CM20 ampere manual reset breakers as installed in the Quaker City trailer.

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For the Sake of Argument

Breaking Mirrors . . .

By Norman G. Shidle

A man looked into a mirror the other day, didn't like what he saw there—and smashed the mirror.

That's the way we all act once in a while, perhaps too often. Seeing around us something we don't like, we lash out at it. Much later, we recognize it as just a reflection of our own thinking or acting or conception.

It's easy to forget that we can't hope to do much about controlling our environment . . . but that we can control what we think about it.

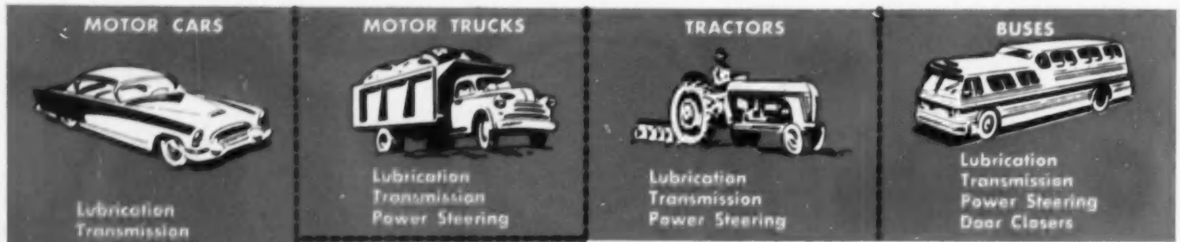
So with the actions and ideas of people we meet. Most often people act just the way we expect them to act. . . . That fellow "who always irritates me" always irritates me. Reason? Partly, at least, because I have predetermined the acts and the ideas with which I am going to be irritated. Whenever and wherever they occur, they automatically push my ready-and-waiting irritation button. I have wired myself carefully and turned on the current for reception.

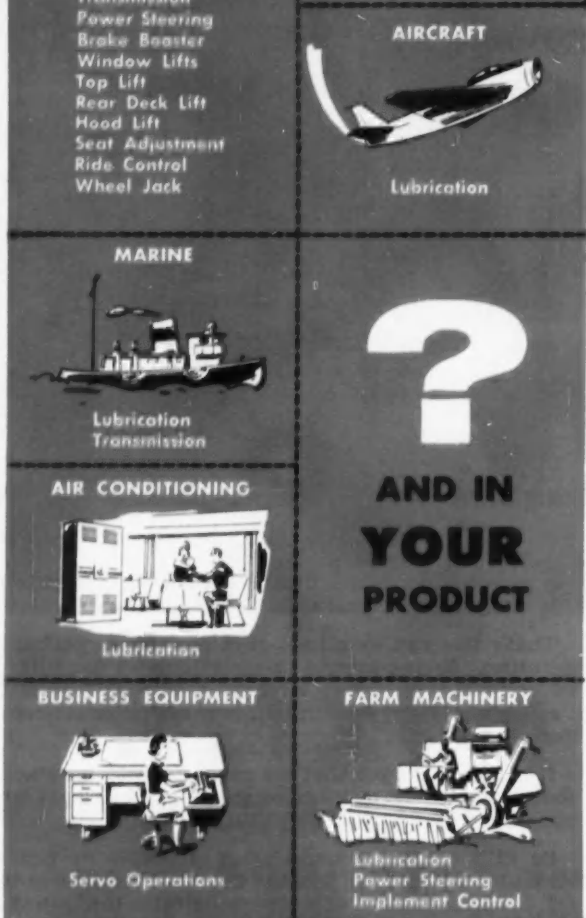
But, it sounds a little foolish to say I go around irritating myself. So, I say instead: "That fellow irritates me." Or, "People who do so and so irritate me."

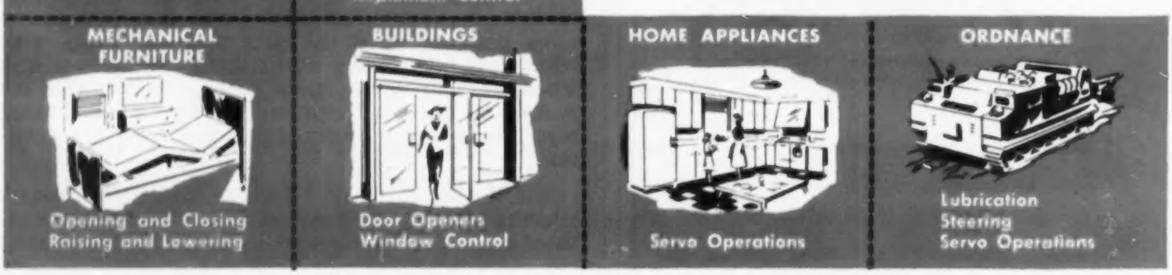
Sometimes we justify—even applaud—our irritation as a sort of sturdy resistance to the lowering of what we see as personal standards. (But others may see them merely as personal predilections or prejudices!)

It *is* easier and quicker to smash the glass than to stop to see the truth about the object which is reflected.

But eventually we may cut our hands . . . or run out of mirrors.







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Hear No Noise

See No Smoke

EVALUATION of reaction kinetics has led to a new concept of combustion for diesel engines. Called the M-system, it clearly shows the following advantages over conventional systems:

1. Virtual elimination of diesel knock, even at very low speeds. Result is a quiet engine with extremely elastic torque characteristics.

2. Less soot formation per pound of fuel at part load.

3. Lube oil contamination no greater than in gasoline engines (because their paths of reaction are similar).

4. It allows true multifuel engines to be developed. These engines can operate on fuels that range from gasoline to jet to diesel fuels without sacrificing performance.

Application of Reaction Kinetics

Reaction kinetic concepts indicate that we can reduce both noise and smoke if we:

1. Limit to a minimum the portion of the fuel involved in autoignition.

2. Allow the fuel to oxidize gradually and try to heat fuel and air together.

3. Mix the fuel with the hot air fast enough to effect a stoichiometric air-fuel ratio before ignition starts and make sure that no more fuel is mixed at any time than can burn with a permissible pressure rise.

These requirements cannot be met with the conventional methods of mixture formation in diesel engines. Certainly, it might be quite possible to follow rule 1 by employing a fine pilot jet for ignition, although nozzle orifices of such a small diameter are undesirable in practice; the real difficulty is in complying with rules 2 and 3 at the same time. Rule 2 calls for caution in bringing the fuel into contact with the hot air, while rule 3 postulates: Mix the fuel with air in successive minute portions, the rate of mixture of each portion being many times faster than that of the preceding one. Furthermore, for those portions of the fuel entering the chamber later in the succession, it seems impossible to observe rule 1, as it makes it necessary to change the properties of the same fuel from highest ignition quality to maximum antiknock rating. But to operate an engine on two fuels, which might be a solution, has never been desirable.

The provision of rule 1 could be fulfilled, we found, only by providing an external source of ignition for the remainder of the charge, or rather an ignition that can at least be timed. A precondition is that an air-fuel mixture be formed which does permit ignition by external means before autoignition occurs, that is, the ignition point of the mixture must be as high as possible. As mentioned earlier, such mixtures are obtained by means of careful heating combined with gradual oxidation as prescribed by rule 2. It is clear that neither can be achieved with a droplet moving in hot air with tremendously high heat-transfer rates.

According to rule 2 the fuel must not be allowed to get into contact with the air in the first stage and therefore it must not be atomized. But as it is to be burnt in the combustion chamber eventually, it must be kept there in a nonatomized form and at a temperature where gradual pre-oxidation and progressive mixing according to rule 3 are possible.

The most suitable spot for this is the surface of the combustion chamber, the temperature of which is sufficiently below the gas temperature to protect the fuel from rapid heating. Thus, at the beginning of mixture formation a process has been adopted whose retarding effect on reaction is well known and dreaded: fuel impingement on the wall. But the only reason this is bad is because reaction has already begun while the fuel was penetrating the hot air. Troubles experienced in conjunction with this latter condition have given the idea of wall

J. S. Meurer, Maschinenfabrik Augsburg Nürnberg

Excerpts from paper, "Evaluation of Reaction Kinetics Eliminates Diesel Knock—M-Combustion System of M.A.N.," presented at SAE Golden Anniversary Summer Meeting, Atlantic City, June 12, 1955. This paper, plus discussion, will be published in full in the 1956 SAE Transactions.

With New German Diesel

contact bad repute, although it was actually applied in engines some 40 years ago.

The need for simultaneous cooling and evaporation implied by the second rule could best be met by spreading a thin fuel film over a maximum area of the combustion chamber. It had to be thin in order to make cooling sufficiently effective right through the fuel film to prevent the top surface in contact with the hot gases from being heated too fast, as a result of the temperature gradient through a thick layer.

Slow and uniform pre-oxidation also necessitates a certain surface area. There is certainly more turbulence in a fuel splash shooting along the wall of the combustion chamber than in a fuel droplet; thus, the fuel particles are partly exposed to the oxidizing effect and partly to the cooling effect of the wall. As a result, the combustion-chamber wall acts as an intermittent carburetor performing a very complex function. The fuel, being already in a highly active state, is still under the control of the wall which, therefore, through its specific properties is well capable of exerting a catalytic effect on the course of the reaction.

Finally, rule 3 has to be complied with. So far the indispensable mixture with the air has not yet taken place and, as we know, it should proceed at an extremely high rate. Without this rapid mixing process, the results of depositing the fuel on the wall to such an extent would be entirely negative with respect to air utilization and fuel consumption. The diffusion rate of the fuel vapor which develops on the wall is much too low to ensure mixing at the necessary high rate. As the only suitable means there remains an air motion which, in view of the application of the fuel in the form of a film, should be well-directed and maintained throughout combustion undisturbed by the latter. It is certainly easier to provide a turbulent air motion by squish, but such a swirl is limited to a short period and breaks down due to expansion of the charge following ignition.

In order to peel off the fuel film in layers, a rotary air motion around the cylinder axis is most suited. It should be produced during the suction stroke and be maintained by the inertia of the air long enough to be still noticeable during the exhaust stroke. The rotary velocity of the air in the decisive phase of mixture formation can be efficiently increased and a favorable distribution of the air in the combustion chamber can be obtained, if the diameter of the latter is kept smaller than that of the cylinder and if care is taken to ensure that a maximum portion of the air gets into the combustion chamber.

However, before we apply such a violent air motion, we must investigate whether we can reconcile it with the requirement of rule 2 for slow oxidation and minimum air contact. Breaking up of the fuel jet occurs as a result of the friction between the jet surface and the air, which is at a minimum when the jet and the air have the same velocity. The nearest approach to this state is made by directing the jet tangentially to the air motion on to the wall. If the jet hits the wall at a distance from the nozzle where it only just starts breaking up, excessive atomization will be prevented, no matter whether it is injected tangentially, against, or in the direction of rotation. If the fuel jet and air move in the same direction, the air will assist in spreading the fuel film.

Selecting the proper distance of the nozzle tip from the wall of the combustion chamber will also help one to comply with rule 1. It has been found by means of high-speed photographs of fuel injection into a hot bomb that the first ignition was always initiated by a small cloud which formed near the nozzle. Obviously, this small fuel cloud develops at the very beginning of injection and, due to the initial impact on the air, consists of very fine particles, the aerodynamic resistance of which is higher than their inertia. These particles, therefore, lag behind the jet. The nozzle tip is located at a point where it is not directly exposed to the rotary air motion, providing a distance between the tip and

the wall which enables these fine particles to separate before the fuel jet hits the wall. It is thereby possible to subject these particles to the normal autoignition of fuel distributed in air. In this manner, the small portion of the fuel which alone, according to rule 1, should self-ignite, is obtained without an additional nozzle orifice. To achieve self-ignition of this portion, advantage is taken of the low ignition point of fine fuel fog formed in the presence of oxygen.

According to rule 3 it is the function of the air motion to produce the mixture so rapidly that at least a stoichiometrical air-fuel ratio is attained before ignition sets in. But in addition, rule 1 requires that autoignition be avoided. The measures so far proposed create favorable conditions to meet both requirements:

1. The fuel, which at the beginning shoots across the combustion-chamber wall at nearly the speed of the air, is considerably retarded in the process and, as the mixture formation period approaches its end, the velocity difference between the rotating air and the fuel increases progressively. In effect, this is the opposite of what happens in engines where the fuel is injected at right angles to the air swirl. There, the relative velocity of air to fuel decreases as the droplet is reduced by evaporation in the course of mixture formation, so that its eventual velocity conforms with the direction of the air motion and it moves along enveloped in its own vapor cloud. Apart from that, each liquid droplet constitutes an undesirable accumulation of fuel which, under engine compression densities, necessitates for correct mixture approximately 6.5 times the air volume required by fuel vapor of equal volume. Once deposited on the wall, the fuel hardly leaves it in the liquid state and, therefore, the risk of wide differences of fuel concentration in the mixture is eliminated.

Fig. 1—One design of combustion chamber incorporating M-system. Hemispherical combustion-chamber is formed in piston. Throat of chamber has recess through which fuel jets reach wall at acute angle.

2. Evaporation of the fuel on the combustion-chamber wall is not instantaneous but is controlled by the wall temperature and by the temperature difference between the wall and the gas. The total kinetic energy stored in the air rotation affects only a fractional amount of the vapor at a time and thus can achieve the prescribed rapid mixture followed immediately by combustion. Thus, the same process that in the conventional methods of mixture formation takes place only once, involving the complete fuel charge, is here successively repeated many times.

In selecting the wall temperature, one must be sure that it is high enough to initiate evaporation, but not so high as to cause thermal decomposition of the fuel molecules to the point of formation of solid soot. Since, due to heat dissipation, the air temperature is lower, initially, near the wall than in the center of the combustion chamber, the desired slow oxidation takes place partly as a surface reaction in the boundary layer between the liquid and vapor phase and partly in the vapor phase.

The evaporating fuel particles quite obviously consist of decomposition products of the first stage (aldehydes, ketones, olefins) that are formed under moderate heat and an extreme air deficiency; these ignite spontaneously at a higher temperature than does vapor formed in the presence of hot air. Consequently, the induction time for these mixtures at the temperatures attained through compression in the diesel engine is now long enough to prevent autoignition within the period during which they are mixed with hot air. Therefore, external ignition and subsequent combustion are possible before the portion of the mixture involved can ignite spontaneously. The instant of ignition is evidently determined by the ignition limit set by the fuel concentration being passed, and the means of ignition are the incandescent fuel particles generated in the limited autoignition according to rule 1. This process is repeated in rapid succession in different places, and at any one time involves only that small mixture fraction which, depending on the rate of evaporation, passes the ignition limit at that time. Similarly to the gasoline engine, where there is no knock if the last portion of the mixture can pass through the flame front before self-ignition occurs, knocking is prevented here by producing, igniting, and burning very small amounts of the mixture at a rapid rate. Thus it is possible to trick the high compression ratio out of its traditional function of triggering self-ignition. This method is also adaptable to the gasoline engine.

What has been described in the foregoing is well confirmed by the engine in operation, although the practical design suffers from a few compromises, especially regarding the removal of the fuel from the wall, which is effected at such a high rate that the increasing gas temperature tends to assist the decomposing reaction.

Combustion-Chamber Design for M-System

There are several ways of translating into practice the rules and instructions given in the foregoing, one design being shown in Fig. 1.

The piston is designed with a combustion chamber in the center forming a little more than half a sphere. The throat of the chamber has a recess through which the fuel jets reach the wall of the combustion chamber at an acute angle. The nozzle has two orifices of 0.015-in. diameter each and its operating pressure is 2500 psi. During the suction stroke, a rotary motion of the air is induced in the cylinder in the same direction as the fuel jets. The rpm of the air swirl is considerably increased when its diameter is reduced at top dead center from that of the cylinder to that of the combustion chamber. The fuel is applied to the wall of the combustion chamber in two splashes, which spread rapidly and eventually flow together to form a continuous film which, as can be seen from the surface appearance of used pistons, covers half the combustion-chamber surface (Fig. 2). Assuming uniform distribution of the film, its thickness can be calculated from the quantity of fuel injected and is 0.0006 in. at full load.

For full load, the combustion-chamber-wall temperature is maintained at about 640 F by an oil cooling arrangement (Fig. 3). A nozzle in the crankcase throws up to 2 qt per min of oil against that side of the combustion chamber where the fuel film is deposited. Cooling of the piston might be effected without oil by suitably dimensioning the cross-sections of the material, but oil cooling makes the piston lighter and also has the advantage that the pistons remain very clean. When the cold engine is started, oil is instantly available at the piston rings and the oil supply of the engine warms up quickly. It is somewhat difficult to control the oil consumption; a very satisfactory solution was found in the present case, however, in the form of wide return channels in the piston skirt.

All prerequisites for the successful adoption of the described system are now fulfilled:

1. Deposition of the fuel on the wall in the form of a film with a small amount of fuel being separated for distribution in the air and initial autoignition.
2. Direction of the jets tangentially to the direction of the air so that the undesirable mixture of the liquid fuel with the hot air is reduced to a minimum.
3. Vaporization in the presence of very little air at low temperature and gradual pre-oxidation as well as rapid mixture of the fuel vaporizing per unit time and a properly timed external ignition.

Although the individual phases overlap, it is possible to make a comparison between a conventional direct-injection engine and the M-system (Table 1).

It can be clearly seen that for the greater part of

Table 1—Comparison of Mixing and Combustion Processes in Standard and M-System Diesel Engines

Normal Diesel System		MAN M-System	
100% of injected fuel		100% of injected fuel	
1) Injection into air	of liquid fuel	1) Injection of liquid fuel	
		about 5% in the air	about 95% on the wall
2) Mixing with hot air		same as in normal Diesel engine (see left)	2) Evaporation at low temperature under lack of oxygen Slow preoxydation
3) Evaporation at high temperature in presence of oxygen			3) Mixing of fuel vapor with air.
4) Autoignition of incontrollable quantities of fuel (low ignition temperature), decomposition to carbon			4) Ignited by incandescent carbon particles produced by autoignition of air distributed droplets
5) Combustion with high reaction rate at the start, increasingly inert toward the end			5) Combustion before autoignition occurs

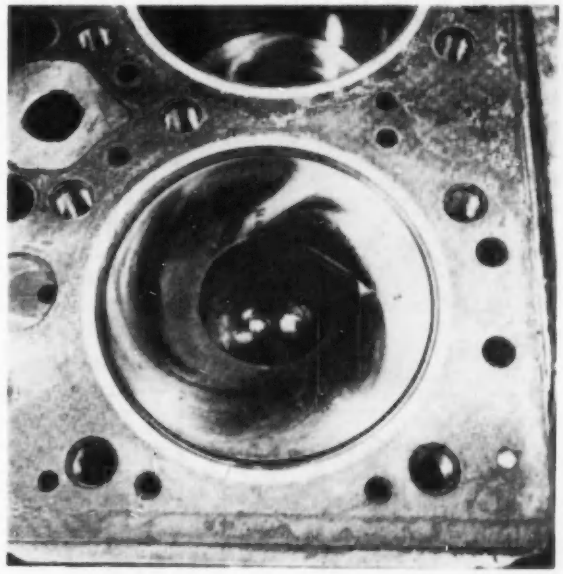


Fig. 2—Appearance of piston top after short running time. (Note that fuel layer has kept clean a strip on combustion-chamber wall.)

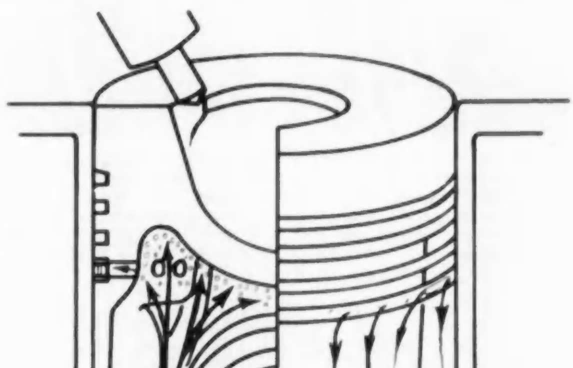


Fig. 3—Combustion-chamber wall is cooled by directing a jet of oil toward piston from below.

the fuel the M-system interchanges points 2 and 3 of the standard diesel system shown in Table 1.

Test Results

The engine test results of the new system are amazing: In particular, typical diesel knock has disappeared over the entire speed range, even during idling and starting of the cold engine. This has been obtained without sacrifice in specific output or in fuel consumption. Moreover, these characteristics are, within wide limits, independent of the fuel characteristics including the cetane number.

The specific fuel consumption of the 6-cyl engine shown in Fig. 4 is given in Fig. 5. The curves of equal consumption extend over wide speed and mep ranges. There is no appreciable narrowing of the spacing in the region of high or low speeds, indicating that there is no definite point beyond which mixture formation is a limiting factor. Note how far the curves extend into the region of low speeds. In road service, these engines show a surprising flexibility and even at 300 rpm combustion is still uniform enough to permit full-load fuel quantity to be injected without trouble. Below 800 rpm, the injection pump, controlled by the governor, supplies an extra starting charge which improves flexibility still more (Fig. 5, full-load curve).

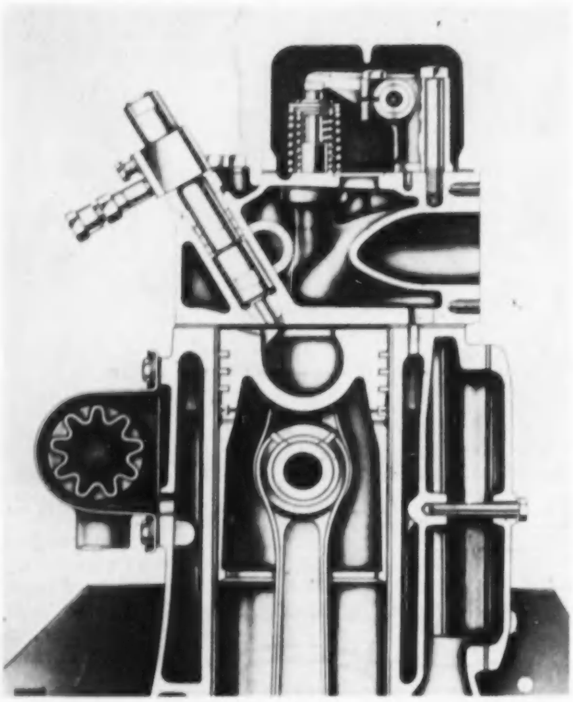


Fig. 4—Sectional view of 4.4 x 5.5-in., 6-cyl in-line production engine. Maximum output (supercharged) is 200 hp at 2300 rpm.

Note M-system has not complicated design. It is essential to locate valves in central plane of cylinder, where they can be designed with maximum cross-section. This is of special importance for intake valve because measures necessary to produce air rotation reduce effective port area.

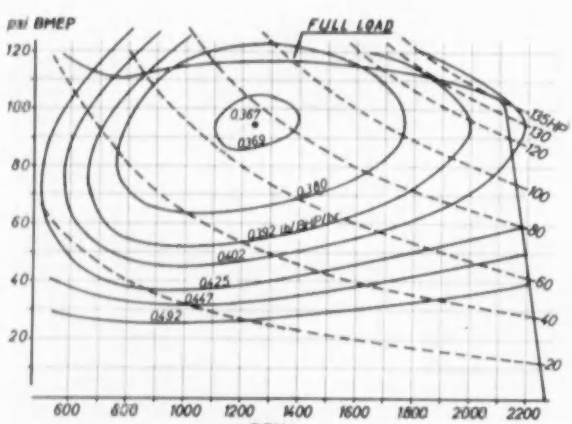


Fig. 5—Specific fuel consumption in relation to engine speed and mep, for 6-cyl engine with natural aspiration. Full-power curve is also shown.

Comparison of Soot Formation

Interesting conclusions with respect to the reactions during mixture formation can be drawn from the amount of soot in the exhaust and the analysis of the used oil. If the discoloration of the diesel exhaust smoke is measured, it will be found that there is a marked reduction when the mean effective pressure decreases, which is readily explained as the result of increasing excess air. However, the influence of the air-fuel ratio must be eliminated if, in order to clarify the mechanism of combustion, we want to find out what percentage of the carbon supplied in the fuel is obtained in solid form in the exhaust. This is done by relating the amount of soot not to the exhaust gas quantity but to the quantity of fuel injected, in other words by making the amount of filtered exhaust proportional to the fuel quantity.

The investigation covered three different methods of mixture formation and the results are recorded in Fig. 6. It is surprising to find that, independent of the air-fuel ratio, a precombustion-chamber engine and a direct-injection engine transform nearly the same percentages of the fuel into free carbon. This clearly indicates that soot formation in the diesel engine is essentially due to reaction kinetic causes. Only in the power range above 75 bmep does the influence of mixture formation become noticeable.

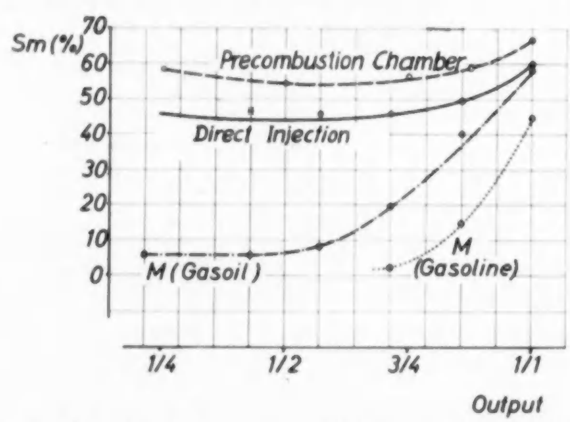


Fig. 6—Smoke in exhaust gases measured for same weight of fuel injected, for several load conditions, at 1400 rpm.

The soot formation in the M-system shows a different trend. In the range up to 50 bmep soot formation is within the limits of a gasoline engine and then approaches the characteristic of the diesel engine because the realization of the M-system in the present design still suffers from some compromises in the region of high mean effective pressures.

This result agrees very well with an analysis of used oils in engines employing the M-system (Table 2). Column 1 gives the data of the direct-injection and M-system versions of the same engine under equivalent operating conditions. It can be seen that the percentage of constituents which are insoluble in standard benzine is much lower in the M-system engine, while the acidity of the oil in that engine is a fraction of that arising in the direct-injection engine. The latter result clearly suggests the different direction of reactions which no longer form acetic acid and formic acid but take place via ketones, olefins, and aldehydes. From the analysis one may calculate the sludge rate, which indicates the amount of benzene insolubles produced in an engine per hp and per mile.

Whereas the value found for orthodox diesel engines is 4.6 oz per 1000 miles and 100 hp, for M-engines it is as low as 1.1 oz per 1000 miles and 100 hp. In gasoline engines it is approximately 1.1 oz per 1000 miles and 100 hp. In other words, the M-engine equals the gasoline engine in low oil contamination.

Cold-Starting Ability

It is unlikely that the fuel, which in the M-system is deposited on the cold wall of the combustion chamber, will vaporize in time to provide quick starting. Only the air-distributed fuel fraction separated from the jet for initial autoignition may do so; but, being very small, the pressure rise attained by its combustion is insufficient to overcome the friction resistance of the engine and so to start the engine on its own.

To remedy this deficiency, two measures have been introduced:

1. The injection of the fuel pump was increased to 180% of the full-load quantity in the range of starting speeds. Consequently, the air-distributed fuel fraction became large enough to ensure sufficient heat liberation after ignition to start wall vaporization. The starting time nevertheless remained longer than that of conventional engines fitted with glow plugs and the exhaust showed unburnt fuel vapor during the first 30 sec.

2. Perfectly reliable and quick cold starting was ensured by means of a small compressed-air burner, which is illustrated in Fig. 7. Compressed air is supplied (a) from the air receiver of the brake system

Table 2—Chemical Analysis of Lube Oil from Standard Direct-Injection and M-System Engines, after Same Service Time

Full Load-Endurance Test Run		
Engine Type	8 × 4.53" × 5.51" 180 HP Direct Injection	8 × 4.53" × 5.51" 210 HP M-System
Hours of Test Run Equivalent Miles	60 2620	60 2620
Neutralization Number	0.3%	0.0%
Insolubles in Benzene	0.8%	0.28%
Specific Sludge Formation	2.3 oz/1000 miles – 100 HP	1.1 oz/1000 miles – 100 HP
Road Test (City Bus Service)		
Engine Type	6 × 4.41" × 5.51" 130 HP Direct Injection	6 × 4.41" × 5.51" 130 HP M-System
Stat Mile	2240	2240
Neutralization Number	0.83%	0.53%
Insolubles in Benzene	1.9%	1%
Specific Sludge Formation	6.0 oz/1000 miles – 100 HP	3.1 oz/1000 miles – 100 HP

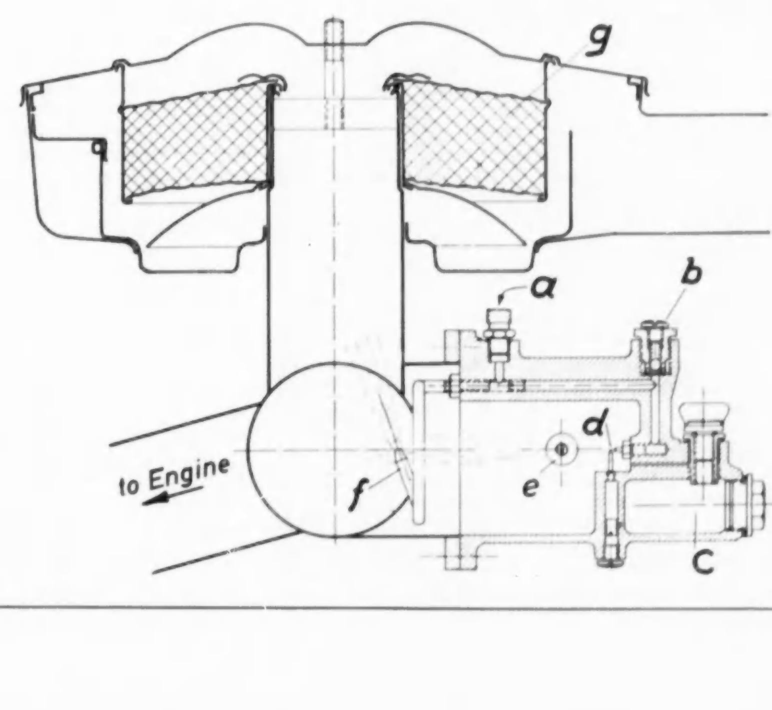


Fig. 7—Cold-starting device in intake manifold: a—compressed air inlet; b—automatic pressure control; c—fuel tank; d—air and fuel nozzles; e—spark plug; f—secondary air nozzle; g—metallic air filter insert.

or by means of a hand pump through a pressure regulating valve (b) to an air nozzle (d) blowing over a fuel nozzle. Some fuel is drawn from the nozzle and, after mixing with the air stream, is ignited by means of a spark plug (e). The small flame developing reaches approximately into the middle of the intake pipe, where it is met by an air stream emanating from nozzle (f) which, being heated by the gases, is discharged through the filter element of the air cleaner into the open. The filter is heated and when, after 20–30 sec, the starter is operated, the engine draws the cold air through the highly preheated filter. Without any of the oxygen of the intake air being consumed, the latter is preheated by about 110 F, which is sufficient to enable the engine to start instantaneously. This cold-starting unit may be operated as long as the engine is warming up because the flame is nearly self-sufficient due to the compressed air supply, and consumes scarcely any oxygen from the intake air. In case the air pressure in the brake receiver is zero, there is a nonreturn valve, which ensures that, during the first revolutions of the engine, the brake compressor pumps the air to the burner before any air is delivered into the brake system. The unit, which operates on gas oil, is controlled by a push-button fitted on the dashboard. With gas oil for fuel, a preheating time of 30 sec, and a starter speed of 110 rpm, the engine will run on its own within 9 sec at –4 F. This arrangement provides most satisfactory cold starting under normal service conditions.

If fuels of a very low cetane number, such as high-octane gasolines, are used, another measure must be taken in addition to the cold-starting device. It is not at all difficult with the above-mentioned equipment to ensure ignition and starting of the engine, but a longer preheating time is required when fuels of low ignition quality are used in order to attain maximum speed and full power. But this difficulty, too, was successfully overcome. The cold-starting ability of a diesel engine is roughly proportional to the amount of fuel which is "air distributed," that is, which is able to undergo autoignition or at least to participate in the preparatory stages. The more the wall application of the fuel can be changed into air distribution, the better will be the cold-starting ability. A first attempt was made by providing an additional third fuel jet directed into the center of the combustion chamber so as to bring about air distribution. The effect on the cold-starting ability of the engine was quite marked. However, under full-load service, that advantage was offset by increased exhaust smoke and rough running of the engine.

Rotating the entire nozzle for low-temperature starts is also very effective, but it is simpler to vary the velocity or direction of the air swirl in the combustion chamber. If the air swirl is reversed by turning the masked intake valve through 180 deg, the jet, being directed against the air stream, will be broken up considerably. Even a reduction of swirl velocity alone improves the cold-starting ability appreciably. A combination of all three measures is most effective and, at a temperature of –4 F, enables the engine to be run up to operating temperature and peak speed within a very short time even when leaded gasolines of 80/86 octane are used.

(Paper on which this abridgment is based is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Excerpts from Discussion

Early Experiments with High-Speed Engine

—**Harry R. Ricardo**
Ricardo & Co., Engineers (1927), Ltd.

DR. MEURER has certainly presented a new line of thought to me, for, from my earliest days I was brought up to believe that, on no account, should the liquid fuel be allowed to settle on the walls of the combustion chamber. As a new line of thought it recalls—and goes some way to explain—the results of our early experiments with compression ignition.

About 1923, when we were setting out to try to develop a really high-speed compression-ignition engine, we chose a sleeve-valve engine on which to start our experiments. Following Hesselmann's early teaching, I believed—and still believe—that an organized air swirl is a highly desirable if not an essential feature if efficient high-speed operation is to be achieved. This we knew we could both obtain and control within close limits with a sleeve-valve engine, and that without any curtailment of breathing capacity. Other reasons for our choice were that with no valves in the head we had much greater freedom of maneuver as regards combustion-chamber shape, and besides, we had become very fond of the sleeve for its own sake.

In these experiments we employed a cylindrical combustion chamber about half the diameter of the piston and about the same height, with a single orifice injector squirting directly downward toward the flat crown of the piston. The spray was in the form of a narrow-angle cone; as a first approximation we sited the injector at about the radius of gyration, far enough out from the center to encounter most of the rotating air but not far enough to impinge against the surrounding walls. This setup, once the rate of air swirl and fuel delivery had been correctly matched, gave pretty good high-speed performance, but with a pronounced diesel knock.

We tried moving the injector about and, somewhat to our surprise, found that we got both the best all-around performance and the least diesel knock with the injector as far from the center as we could get it, namely, only about $\frac{1}{8}$ in. from the side wall. Under these conditions nearly half of the total fuel injected must have impinged against the cold side wall, as indeed was proved by motoring the engine and, for a few cycles, injecting a noncombustible pigment. Unconsciously, then, we were

working under somewhat the same conditions as Dr. Meurer advocates.

Some years later, when our sleeve-valve diesels were being marketed in the Far East, we were faced with the problem of getting them to run on a very low-cetane fuel from Borneo. To this end we fitted inside the combustion chamber a heat-insulated austenitic steel liner. This proved very effective, but we found it necessary also to speed up the rate of air swirl greatly, to the best of my recollection, from about 9/1 to 13/1, presumably because of the greater rate of evaporation of the liquid fuel plastered onto a much hotter surface.

Recently we began developing a technique of combustion photography using a wide variety of combustion-chamber shapes, including direct injection, "Comet" type, "Whirlpool," and others. In all of these we have been a little surprised to find how considerable a portion of the fuel is, in fact, deposited on and burnt from the surface of the combustion-chamber walls, but this is an accidental, not a deliberately chosen condition, as in the M.A.N.

In short, I suggest that the combustion conditions that Dr. Meurer advocates are present in much lower degree in every high-speed solid-injection diesel engine. To Dr. Meurer and his associates is due the credit for recognizing, evaluating, and cultivating them.

Air Cargo . . .

. . . will never realize its potential until costs can be lowered. Turboprop-powered airplanes can make possible a sharp reduction in ton mile rates.

Based on paper by **Carlos Wood,** Douglas Aircraft Co., Inc.

AIR cargo lines appeared by the hundreds after World War II, when ambitious returning fliers went into business with surplus C-46, C-47, and C-54 aircraft. Today they are washed up and there are left only such independent operators as Slick, Flying Tigers, Seaboard and Western, and Transocean. And these have to depend upon maintenance, manufacturing, aircraft leasing, and Korea to stay alive.

Scheduled airlines have moved into the cargo field to some extent, in fact, air cargo business has grown steadily, but at nowhere near the rate anticipated. What is the reason for all this?

Basically, the reason is the relationship between the rates that had to be charged to stay in business, and the rates at which cargo would move. Fig. 1 shows the direct operating cost per ton mile versus airplane capacity in ton miles per hour, together with the approximate corresponding cargo rate. Solid lines represent existing airplanes; dotted lines represent estimates for possible turboprop-powered airplanes. Here is the field for the turbine propeller engine with its high take-off power, low fuel consumption, long range, and large lifting ability.

Results of a study into the possibilities of air cargo volume versus rate per ton mile are shown in Fig. 2. Hauls of less than 400 miles were not considered. Here reduction in rates to about one-half their present level multiplies volume by 10.

Fig. 3 attempts to put a time scale on the possible

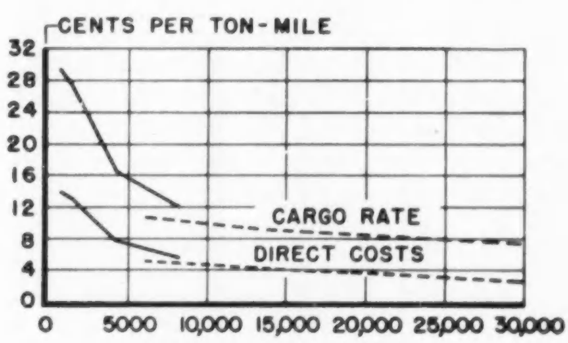


Fig. 1—Solid lines show cargo rate and direct costs with existing airplanes; dotted lines show what happens (estimates) when turboprop-powered aircraft are used.

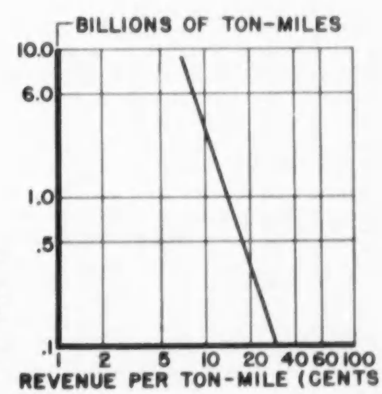


Fig. 2—Here is a study to show what might happen to cargo if rates were lowered. Air hauls of less than 400 miles were not considered in the study.

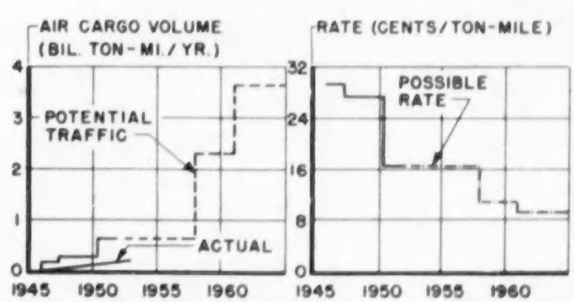


Fig. 3—Potential cargo volume at various rates plotted on a time scale. This is what could happen, not necessarily what will happen.

variation of air cargo rates and shows the actual and possible cargo volumes that might result. It takes large capacity airplanes for low cargo rates and high cargo volume, and this fits in with large turboprop engines which can be developed from existing highly efficient jet engines.

One cannot over-estimate the importance of high capacity cargo transports in our world military picture. Rapid movement of military forces, equipment, and supplies on a world-wide basis is essential if we are to maintain peace and our world position without economic disaster brought on by the need for the excessive forces that would be required without such mobility. (Paper "Present Trends and Future Possibilities in Air Transportation" was presented at SAE Golden Anniversary Annual Meeting, Detroit, Jan. 10, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Jet Transports . . .

. . . can be competitive and play a major role in the transportation system. In fundamentals the jet is substantially better than existing aircraft.

Based on paper by **George S. Schairer,** Boeing Airplane Co.

JET airplanes provide airline speeds from 50 to 100% greater than existing equipment around which airline cost analysis procedures have been developed. Such a large change in one variable calls for a review of the basic cost formulas since they were never intended to be usable over such a large range of variables. For this reason, the principal problem to be faced in studying the economics of a jet transport is the effect of speed upon the analysis.

When such a study has been made and basic data have been assembled, it can be seen that to a first approximation, the airplane cost to provide a unit of capability will be proportional to the operating weight empty of the airplane required to carry a given amount of payload. Fig. 1 shows payload carried per pound of operating weight empty versus range, and ton knots of payload carried per pound of operating weight empty. Similar data are plotted for typical modern reciprocating engine powered transports. Here is a measure of work capability of the airplanes per unit of investment cost. It will be seen that the jet makes a very creditable showing of work capacity per pound of weight empty as compared with existing transports. Fundamentally, the jet transport is substantially better.

It may be countered that there is more to it than that. Existing aircraft have already been built in large quantities while the jet is a new airplane starting at the top of its learning curve with high initial development cost. However, using the best available dollar costs at current market prices, a measure of passenger nautical miles per hour per dollar first cost has been made, and is presented as a comparison of the Boeing 707 jet transport with existing production reciprocating-type airplanes (Fig. 2).

Despite the necessarily high costs of a new airplane, the investment required per unit of passenger hauling capability is no greater than that being made in existing transports where development costs have been written off and the airplanes are well down their learning curve.

The jet transport can start with equal advantage. As it is procured in quantities in subsequent years, it will show very marked improvements in invested cost per unit of carrying capability. The fact that this can be so is most reassuring. (Paper "Economic Considerations of a Jet Transport Airplane" was presented at SAE Golden Anniversary Annual Meeting, Detroit, Jan. 10, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

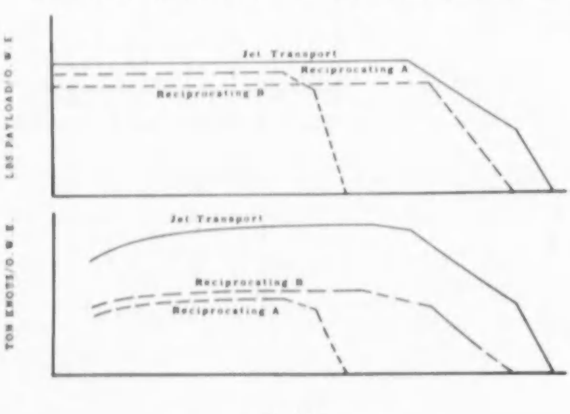


Fig. 1—It is only when the jet transport's speed is taken into full account that a true measure of its cost can be reached. Here a jet transport is compared with reciprocating engine types in work capability per pound of weight empty.

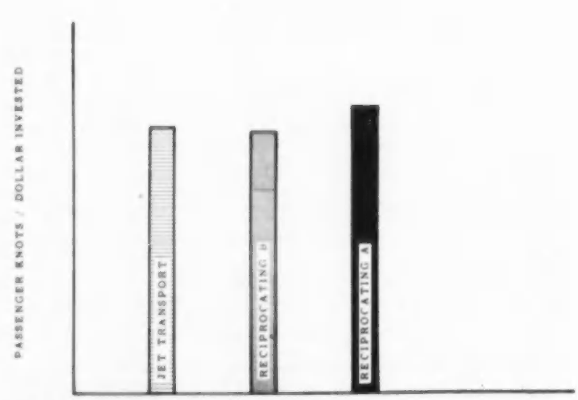


Fig. 2—Passenger knots per dollar invested puts the Boeing jet transport in a very favorable light as contrasted with existing reciprocating engine transports. This chart, which shows fleet investment for equal work rate, indicates the jet can start out on an even basis and make improvements in the future.

What goes on Inside the

Double Disc Brake

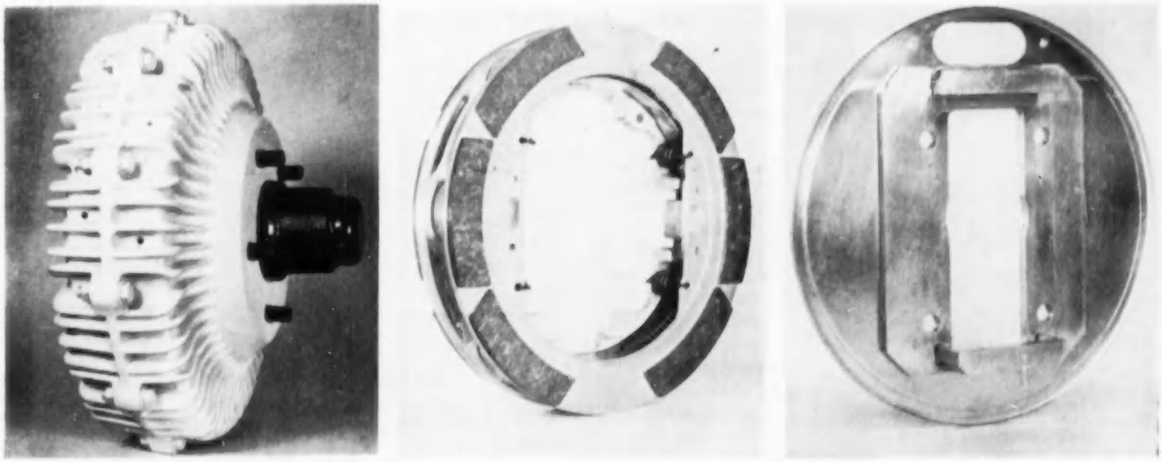
D. A. Gotsch, Auto Specialties Mfg. Co., Inc.

Based on paper "The Auto Specialties Double Disc Brake" presented as part of the panel **New Approches To The Solution of Brake Problems**, at the SAE Golden Anniversary Passenger Car, Body, and Materials Meeting, Detroit, March 3, 1955.

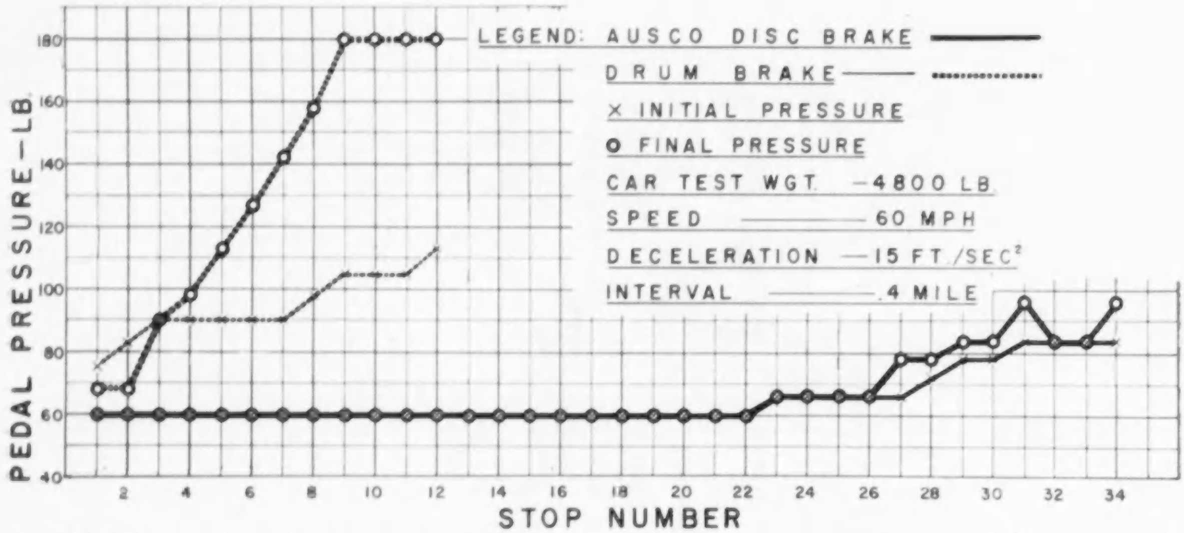
THE DOUBLE DISC BRAKE manufactured by Auto Specialties Mfg. Co., Inc., is their solution to the problem of brake fade. Its light weight "finned" aluminum housing permits heat to dissipate rapidly. And a ball and ramp method of self-energizing gives plenty of braking power and smooth braking with little pedal effort.

Repeated brake applications, or continuous use as when going down a long hill, causes some shoe and drum brakes to overheat. The drums can become so distorted and warped that the shoes cannot contact them. The heat also impairs the friction qualities of the lining. A double disc brake evidently does not warp from the heat because of its ability to transfer internally generated heat quickly to the surrounding air. Also, unlike drum brakes, the lining contacts the friction surface uniformly. This gives more effective braking surface and longer lining life.

The disc brake operates on the principle that



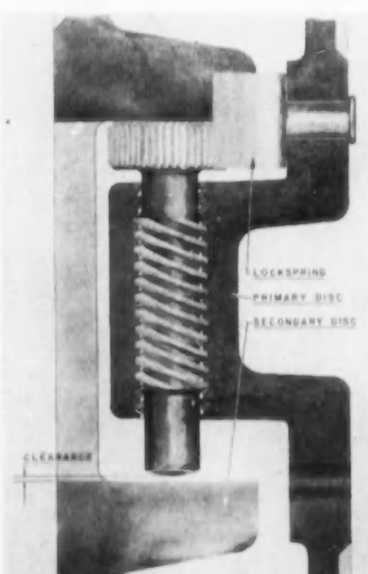
BRAKE HOUSING (left) is made of die cast aluminum with ferrous alloy insert rings on the inner sides which comprise the friction surface on which the brake lining material bears. Organic lining material is bonded in segments on the outside surface of the actuating discs (center). The brake mounting (right) consists of a stamped steel splash shield and steel torque plate. The inboard actuating disc anchors on the lugs of the torque plate and the outboard disc floats freely on six balls and is kept in position by return springs.



FADE TESTS on an 11-in. drum brake showed that at 60 mph the foot pedal force necessary to stop varied from 68 lb on the first stop to 180 lb on the twelfth. Pedal effort on the disc brake remained at 60 lb for the first 22 consecutive stops and varied to a high of 96 lb on the 34th stop. Not only do disc brakes require less pedal pressure, but they do not fade as quickly as drum brakes. Both considerations are vital to safe operation.

when a spinning disc is brought into frictional contact with a stationary disc, the spinning disc will be brought to a stop. The double disc brake has two pressure plates on which friction brake lining material is bonded. Braking occurs when the two pressure plates are pushed away from each other and forced into contact with the inside races of the inner and outer brake housings.

This is accomplished as follows: When the foot pedal is depressed it sends hydraulic fluid through lines into the chamber of the annular piston assembly. (See sketches below.) The o-ring pushes against the annular piston and forces the actuating discs (or pressure plates) apart. The brake lining on the discs then contacts the ferrous friction surface of the rotating housing. This turns the outer disc a few degrees relative to the inner disc. (The inner disc is anchored to the lugs of the torque plate, while the outer disc floats freely on six steel balls.) The steel balls are made to roll up inclined seats and force the discs even farther apart. Thus additional force is applied to the friction braking surface.

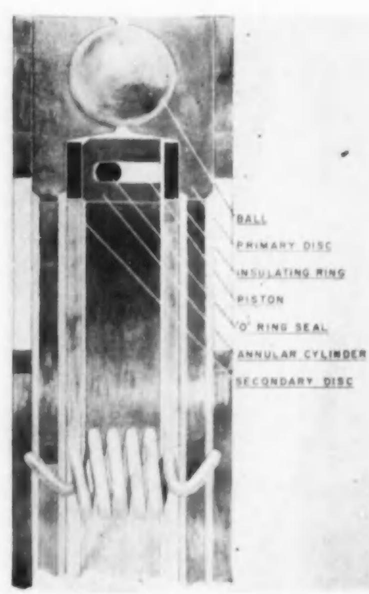


◄

AUTOMATIC ADJUSTER compensates for lining wear. There are two adjusters per brake, positioned between two lugs in the secondary disc. When brakes are applied the discs move in opposite directions. If the linings have worn the screw contacts the bottom lug before the linings contact the brake housing. This pushes the screw through the boss and a spring clip locks it from returning. When brakes are released, head of the screw contacts the top lug and prevents the discs from moving all the way back into their original positions, thus compensating for lining wear.

►

ANNULAR CYLINDER ASSEMBLY between the discs actuates the brake. Cylinder groove is sealed with an o-ring which bears on an annular piston. Bakelite rings are used to restrict heat flow to the hydraulic fluid. Cylinder areas can be varied to produce the desired hydraulic balance, front and rear.



When the foot pedal pressure is released, coil springs pull the plates back toward each other.

Since the ball seats are inclined plane surfaces and the balls follow a straight line path, the energizing force is always directly proportional to pedal pressure. So, stops are smooth and under control with no "wrap-up" at low speeds, and at high speeds the tendency to lock is reduced.

Double disc brakes have relatively few parts which makes for easy assembly. Ausco's brake has a lining area of 47 sq. in. The outer diameter of the friction surface is 11.5 in. and the inner diameter is 9.5 in.

A single lever mechanical emergency brake is incorporated in the rear brake assemblies. The lever is attached to both primary and secondary actuating discs with pins tapered in opposite directions. As the linings wear and the discs move further apart, the lever maintains the same original position, which makes it self-adjusting. Movement of the lever rotates one disc with respect to the other, causing the balls to roll up the ramps and energize in either the forward or reverse direction.

Comparative durability tests were run on cars equipped with 11-in. drum brakes and double disc brakes. It was found that disc brakes have approximately 100% more lining mileage than conventional 11-in. drum brakes.

(The complete paper on which this article is based is available together with three other papers presented at the panel New Approaches To The Solution Of Brake Problems. This publication, SP-138 is available from the SAE Special Publications Department. Price: $1.50 to members, $3.00 to nonmembers.)

Guided Missiles . . .

. . . require close control to be effective. The more effective they are, the greater the cost and complexity. But complexity lowers reliability; hence the best weapon represents a compromise.

Based on paper by **H. A. Eubank, C. C. Ross, and R. C. Stiff,** Aerojet-General Corp.

THE operational reliability of a guided missile powerplant is of prime importance. If the powerplant fails, so does the missile flight. Close power control from the rocket may be required and, if the missile follows an unpowered ballistic trajectory, the power and total impulse control can be of great importance. For example, a 1% variation in specific impulse on a 1000-mile ballistic missile will cause the missile to miss a specific target by as much as 21 miles, assuming the controls and guidance are otherwise perfect.

Guided missiles will have an imposing array of electronic equipment and some of it will be subject to damage under vibration. A thrust oscillation of ±3% is considered generally to be satisfactory for operation of a liquid propellant rocket motor; at 10,000-lb thrust, this is a variation of 600 lb. If the oscillation frequency is about 100 cps, the missile is subjected to 600-lb blows 100 times a second. The effect of this force on missile acceleration is small, but it may create a serious problem with some components located near the powerplant or affixed rigidly to it.

In the case of a liquid propellant rocket, the attainment of high reliability is complicated not only by power control problems and vibration, but also by environmental operating requirements and weight economics.

After performance tolerances and environmental conditions have been established, specific reliability objectives must be determined. Here is a statement of the problem:

1. Missile effectiveness will depend on how close the missile comes to the target, or the ratio of hits to the number launched.

2. The greater the percentage of hits, the more complex and expensive the control system and other rocket components.

3. The more complex the control system and rocket components, the lower the reliability of the missile.

To develop the best weapon, compromise is unfortunately necessary. The final design is not as reliable as the statistician wants, nor as simple and high performing as the engineer desires, but the economist is pleased because it shows the greatest target effectiveness per dollar expended. The following example illustrates the magnitude of this consideration:

If the missile and control system required to hit the target 90% of the time under all conditions costs $1,000,000, and a missile which gives 50% hits costs $300,000, consideration should be given to using several vehicles having the lesser reliability. Three missiles of 50% reliability would give an 87.5% assurance that at least one missile would hit the target, while four missiles of 50% reliability would give a 94% assurance of success.

(Paper "Some Aspects of Guided Missile Powerplant Reliability Requirements" was presented at SAE Golden Anniversary Aeronautic Meeting, New York, April 20, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members; 60¢ to nonmembers.)

Mechanized Manufacture Improves Reliability of

RELIABILITY of performance and mechanization of production as applied to airborne electronic equipment are dominant themes of the electronics industry today. Actually these are one and the same theme, because the purpose of mechanization is to produce more uniform and more reliable electronic equipment for airborne use.

In a piloted aircraft, the proper functioning of all its electronic equipment is essential to the success of practically every flight. It is true that a pilot usually can take over and bring the plane down safely in the event of an electronic failure, but this generally means an unscheduled termination of a commercial flight or an aborted mission in military service. The modern high-altitude high-speed plane needs its electronics more and more each year.

In unpiloted aircraft, the situation is much more critical. Failure of just one tiny component, wire, or joint out of the many thousands in its electronic brains and controls will invariably mean missing the target. If the target is an enemy aircraft carrying an atomic bomb, the possible consequences of such a malfunction are staggering to contemplate.

How to Achieve Reliability

Reliability begins with components. Because the reliability of a complete electronic system is the product of the individual reliability percentages of the components, perfection would appear to be just about essential in any electronic unit having over a thousand parts. Production costs skyrocket, however, with each additional approach to perfection. Fortunately, in practice the laws of probability and statistical theory do not work out quite as pessimistically as predicted by mathematicians—chiefly because the ingenuity of man can fix a lot of things in old-fashioned ways that still get results.

At least one component manufacturer is meeting reliability requirements by setting up a separate manufacturing section where reliability is the dominant theme. Whereas ordinary production workers generally are trained in the philosophy that it costs less to fix or reject after building, these special workers are given perfection as their only goal. Many additional inspections and tests are incorporated in the production setup, even including x-ray inspection of internal joints in finished components and x-ray inspection of soldering on terminals. As a result, a reliability record of one failure in 100,000 has been achieved in a component produced for use under known environmental conditions.

As another example, tubes for proximity fuses are being produced with a reliability of 99.99% plus, in plants or departments where management is quality-minded. Nevertheless, cost of these tubes is comparable to that of broadcast receiver tubes—a proof that the proper attitude throughout a plant can give reliability along with low cost.

Where there is a will, a way can be found to make entire complex electronic systems reliable as well. Thus, the vacuum-tube repeaters inserted in submarine telephone cable every 20 miles are designed and built for *no failures* in 20 years.

Environment is becoming increasingly important

VIA SILK SCREEN, these girls at RCA television plant apply chemical resist to copper-clad laminated boards. Boards go through oven in foreground to bake on resistant coating. Then etching compound eats away areas not protected by the resist, leaving the desired etched wiring.

Electronic Equipment

John Markus, Associate Editor, **Electronics**

Based on secretary's report of Panel on Problems in the Manufacture of Electronics Equipment held as part of the SAE Production Forum at the SAE Golden Anniversary Aeronautic Meeting, New York, April 21, 1955.

as a factor affecting reliability of electronic equipment. Where this can be specified in accurate detail, the component manufacturer has a much better chance of producing units that will stand up in use. As we approach the thermal barrier in supersonic aircraft and plan for altitudes up around 100,000 ft, the old rules for rating and derating components must be scrapped.

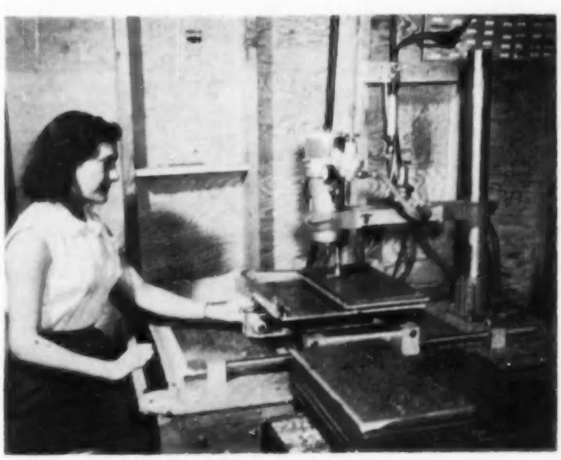
Ten years ago it was assumed that a 20 F rise in ambient temperature would cut the life of a component in half. We now know that the relationship of reliability to temperature is far more complicated than this. A rise in temperature causes a lot of little nuisance failures, even though the temperature rating is not exceeded. Thus, with a nominal temperature limit of 105 C it is still highly desirable to keep that part down around 85 C when reliability and long life are paramount factors.

A comprehensive reliability test program, aimed at the detection and elimination of design weaknesses long before the first missile is fired, is essential. Such a program will guarantee a rapid rise in reliability. This is the way to go—by designing and manufacturing reliability into systems, not by legislating or inspecting for it.

Learning From Failures

There is a cause for every failure. The discovery of these causes is becoming a science in itself, as exemplified by the outstanding detective work on the Canberra jet transport failures. In the electronic field, feedback of failed components to the manufacturer for analysis is revealing some surprising causes, invariably traceable to product design. Internal hot spots are one such cause, heretofore unsuspected because the external ambient was well below specifications.

Another new environmental factor is the unsuspected effect of organic materials, particularly in



HOLES CAN BE DRILLED one at a time in etched wiring boards with this DuMont Pantodrill. The operator moves the stylus along grooves in the template. Each time the stylus drops into a template hole, the air-powered drill comes down to drill a corresponding hole in the board.

FOR LONGER RUNS, holes can be punched in etched wiring boards with one-shot compound perforating and blanking dies, as in the plant of Methode Mfg. Corp. in Chicago. The huge presses are well suited to long production runs, where usage justifies set-up cost.

hermetically sealed devices. As one example, high resistance films can develop on sliding contacts in potentiometers, where there should be zero resistance. This action in the gyro of an autopilot can mean that a pilot might suddenly find himself flying upside down. In the contacts of relays, and in many other components even the lowest level of corrosion on metals or organic film formation can be disastrous. Vapors released by new insulating materials in the confined atmosphere of electronic equipment also have caused severe electrical erosion. The slow deterioration of organic insulating varnishes used in magnetic cores yields products which can destroy other normally protective varnishes leading to rusting of the rotors in selsyns, for example.

These environmental effects were unpredictable beforehand, because of the time and expense that would be involved in testing each new material under every conceivable combination of circumstances. The solution, rather, lies again in high-speed feedback of information from the field, so that causes of failure can be determined and cured as rapidly as possible.

If a fault can be described as mechanical when a part is replaced without changing its electrical rating or circuitry, then around 80% of the trouble with electronic gear is mechanical. It may be either an internal or external flaw, and may be due to vibration, shock, deformation, cold flow, overheating, or a host of other causes not involving circuitry. Misapplications of components, resulting in over-voltage or over-wattage, are true electrical failures and account for a large part of the remaining faults.

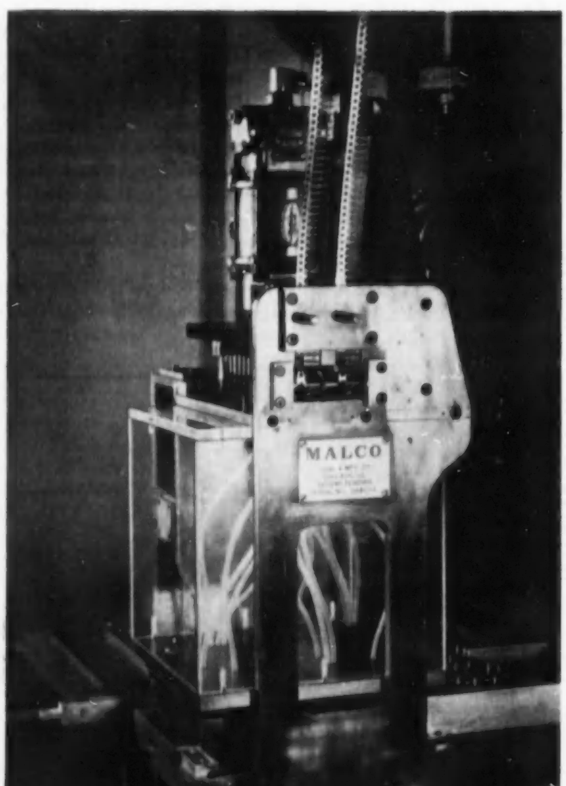
Engineering mistakes tend to repeat themselves in cycles of about 10 years. One reason today is that so many of the engineers who designed World War II equipment have gone up into management, and the younger engineers do not have first-hand knowledge of how the equipment is used and abused in the field. Too often the young fellows feel that they must develop new circuits and use untried techniques that get them in trouble. It is generally better to use simple field-tested circuits then to invent new circuits.

Choice of Flux is Still a Problem

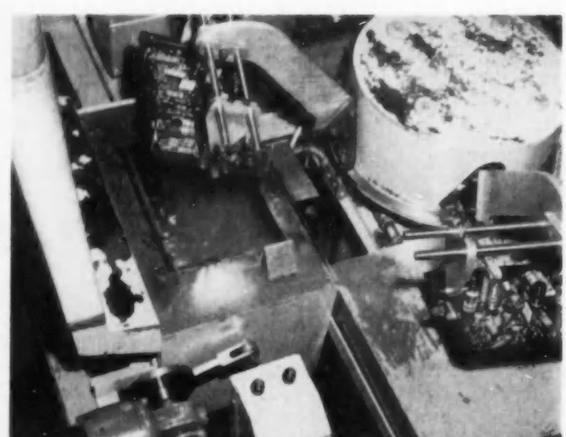
One large manufacturer of military electronic equipment asserts that it has found no truly non-corrosive activated flux. This firm uses only rosin. Here there is considerable difference of opinion in the industry. Some justify the use of corrosive flux on the basis that the heat of soldering produces complete decomposition to an inactive product. Conversely, others insist that the act of soldering entraps some of the flux in such a way that it cannot be washed out by any known solvents. Furthermore, some insulating materials liberate corrosive products when heated to soldering temperatures.

With dip soldering, reducing immersion time to a minimum reduces damage to the phenolic base. In general, the molten solder in a dip soldering setup is contaminated far more rapidly than is commonly suspected. The pot should therefore be emptied more frequently for a fresh start with new solder.

With etched wiring, oxidation of copper during storage can make soldering difficult. When silver is used over the copper for protection, however, it must usually be protected to avoid sulfide tarnishing.



CHAIN-FORM TERMINALS unwind from two overhead reels. This machine, built for Admiral, cuts them off and drops them down flexible plastic tubes into holes in the etched wiring board below. As many as 40 terminals can be installed in one operation in any desired positions.



AUTOMATIC DIP-SOLDERING MACHINE swings wiring board at left from position where it is sprayed with flux to pot of molten solder at rear. Board then floats on molten solder for about 8 sec. Mechanism in foreground oscillates spray guns.

The use of a tin alloy electroplate to produce an easily soldered surface is broadly useful.

Ultrasonics Doesn't Help Dip Soldering

Ultrasonic generators have not as yet proved to be the answer to reliable dip soldering. One reason is that ultrasonic generators powerful enough to activate a large solder pot are not yet commercially available. Another factor is the overeffectiveness of ultrasonic cavitation, in that it removes contamination products from the phenolic base at the same time that it removes oxide from the etched wiring. It has also been found that life of the ceramic solder pot itself is short when the solder is being vibrated at an ultrasonic frequency.

Silver Migration is Not a Serious Problem

Silver migration, currently in the technical news, is actually not too much of a problem to most electronics manufacturers. It can occur in a high-humidity environment when there is a steady dc voltage in conjunction with an organic plastic, but the time required for this is generally beyond the normal life span of airborne electronic equipment. It is more serious in transoceanic telephone cable repeaters, which must operate underwater for a minimum of 20 years, and in ordinary telephone equipment which is built for a life expectancy of 40 years.

Fresh Hardware Solders Better

Terminals and hardware intended for soldering should be used as soon as possible after fabrication. One manufacturer has found that even plated terminals can be stored only up to three months and still retain easy solderability. By applying a rust-inhibiting film and then storing in tote boxes lined and covered with vapor proof paper, this period can be extended up to one year for plated brass terminals.

Embedment Resins Give New Protection to Components

In the past, hermetic sealing was used extensively with the hope of producing favorable environment for electronic components, or rather to isolate the component from the unfavorable environment in which it was used. This went a long way toward solution of the environment problem but its drawbacks were high cost, bulkiness and inaccessibility for repair plus troubles with organic vapors within the container. The trend today is toward use of embedment resins to replace hermetically sealed containers. This is really a modification of the old potting technique wherein a hot bituminous material was poured over a transformer in a can. Now the can is eliminated, the embedment material itself serving as the housing.

There is no one combination of resins and accelerators that is satisfactory for all applications. A choice must be made from among a considerable variety of new materials now available.

With embedment resins, differences in the coefficient of expansion of the resins and the component materials can cause cracking and voids during heat cycling. One way of overcoming this is to use 325-mesh (44 micron) glass beads as a filler in the resin. Another is by first applying a somewhat flexible coating resin to the components either by dip or spray.

***SERVING* on the panel which developed the information in this article were:**

E. M. Wise, panel leader
International Nickel Co.

John Markus, panel secretary
Electronics, McGraw-Hill Publishing Co., Inc.

L. M. Clements
Crosley Division, AVCO Mfg. Co.

Hugh Johnson
Grumman Aircraft Engineering Corp.

M. R. Johnson
General Electric Co.

R. C. Miles
Airborne Instruments Laboratory, Inc.

R. E. Moore
Western Electric Co.

J. J. Preisler
Sperry Corp.

How to Meet Military Production Deadlines

Although haste and reliability are not compatible, great strides have recently been made in producing good equipment in a hurry to meet new military needs. One large manufacturer starts assembling the production team the day that the first design print emerges from the laboratory. This team starts making parts right then, even to the extent of making molding and punching dies. Their goal is to keep right up with the output of the design department. Dies are hardened right from the beginning, on the assumption that these probably will be correct even though the rest of the system has not yet been designed. Experience has been that three out of four of these dies are usable when production on the final design begins to roll.

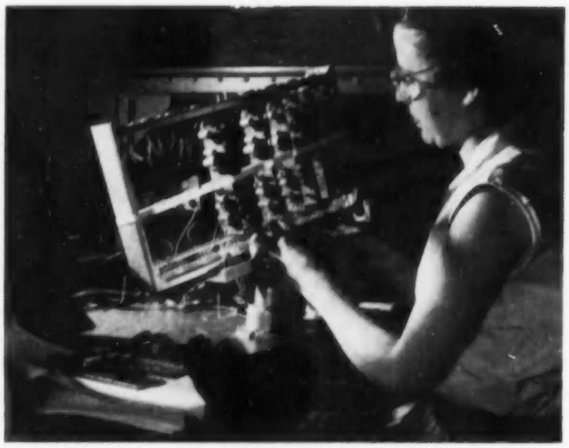
Initial production of samples in this way reveals any need for design changes, and these are encouraged because they do not hold up production at this early stage. Many good ideas come back to the design department from the shop people who build the first model. The average issue of drawing is 3 per component.

It has paid off to bring the production people in early who have to suffer and sweat over the design later. With this technique it was possible to start a completely new airborne electronic bombing system and roll it out of the door in 16 months.

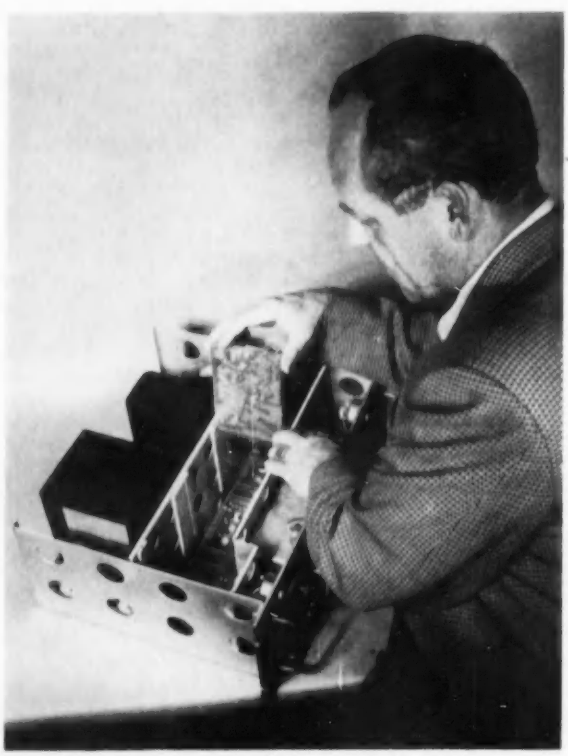
Living With Design Changes

Changes are a bugaboo for the entire life of an electronic system. It has been said if the Air Force were to put all its maintenance people to work making all of the changes suggested by manufacturers of their electronic equipment, the job would take a fantastic number of years. However, it was pointed

out that out of 100 such changes only five or six are really essential for curing major faults. It was recommended that manufacturers concentrate on getting the Air Force to select these. One reason for changes is the tactical need to take the equipment up higher than the altitudes for which it was originally designed. In most instances the manufacturer must foot the bill for design changes, to protect his reputation and to avoid the costly red tape involved in getting government authorization for



WIRING BOARDS ARE INTERCONNECTED by hand, usually, after components have been inserted in the boards automatically.



NEW WAY TO INSTALL WIRING BOARD is to design equipment so that board pushes down vertical slides into special connector at bottom of chassis. Boards of this type are easy to replace.

such extra charges. Initial bids should therefore take into account the expected cost of feeding back failure data and making needed corrections.

A practically unanimous complaint of electronics manufacturers is that at present the military does not provide sufficient time and money to probe into the reliability of components and systems. Manufacturers feel strongly that they could make much better equipment if methods of awarding bids gave fair cognizance to these factors and to the reputations that firms have achieved for producing reliable equipment. In government procurement procedures it is extremely difficult to reward a good manufacturer by placing a dollar sign on longer life—and upon human life.

In the long run, an investment in greater reliability pays for itself many times over. A Navy investigation revealed that maintenance of most of its airborne electronic equipment over its useful life cost between 10 and 100 times its original price. Truly here should be an ample source of funds to pay for ruggedized long-life components, if a means can be found for transferring maintenance appropriations to initial-contract funds.

Transistors Help the Cooling Problem

Transistorizing is another technique for getting airborne electronic equipment out of an unfavorable environment. Replacement of tubes with transistors permits such radical miniaturization that in many cases the equipment can be mounted right in the cockpit, perhaps directly behind the associated indicating meter on the flight panel. The pressurized and sometimes cooled human environment of the cockpit is much easier on electronic components.

The Dead-on-Arrival Problem Stirs Up Controversy

The advisability of a 50-hr run-in of electronic equipment at the manufacturing plant has been considered. About 85% of the radar equipment was inoperative by the time it reached one aircraft manufacturer. Facetiously it has been pointed out that this could be the fault of the Navy inspector, who jarred the equipment when he put on it the Navy stamp of approval.

Run-in would reveal such glaring assembly defects as an input power lead soldered directly to ground (which actually was found in one dead-on-arrival radar!), along with design defects. On the other hand, run-in would shorten the life of the equipment and thereby result in a whole new family when the equipment was eventually installed in an aircraft.

There is a positive correlation between failure of components in storage and in the field. Quality cannot be achieved solely by inspection; it must be designed into the equipment and into its components. Early failure of equipment corresponds to infant mortality, while later failure corresponds to old-age mortality.

Failure of electrical people to recognize the importance of vibration damping, strength of materials, and the other tools of the mechanical engineer is the cause of another large group of electronic faults. Even the basic fact that a triangle is a rigid structure is being overlooked by young engineers.

With many components of vibration frequency

and amplitude being unknown until the equipment is in actual use, the rather desperate smash-and-repair technique is one of the few available means of building vibration-proof black boxes. This involves operating the unit on a shake table until something breaks, then repairing, beefing up, boosting the shakes another 5 g, and repeating until it is no longer possible to produce failure over the frequency range of 75 to 200 cps.

Dip embedment has been used successfully over entire equipments as well as over components to improve resistance to vibration. In another instance the same result was achieved by applying several coats of polyethylene dope over everything.

Opinion is divided on whether to put on shock mounts or to bolt down the black boxes. The goal today is to keep the plane dense while still achieving good ventilation and accessibility. As a result, the trend is toward fewer packages in a system. This cuts down the space available for shock mounts and for connectors.

Connectors, incidentally, are also in a state of transition. Performance is being improved by applying potting material around the leads after assembly, but increasingly high altitudes are introducing new problems that may necessitate going to larger connectors with more clearance between pins.

Mechanization Requires New Thinking

Mechanization of electronic production requires a still higher quality level of incoming parts, because as yet the machines do not know how to throw out parts which are just a little bad. Human operators, on the other hand, do this instinctively with scarcely a break in their working cycle. This is not a serious limitation of mechanization, however. It is sometimes cheaper and equally feasible to fix one or two parts of a system after it is built than to give 100% qualification tests of components beforehand.

Mechanization imposes two new disciplines on electronic manufacturing—the need for standardization and the need for further cooperation between the design engineer and the manufacturing technician. In return it gives a rewarding uniformity in quality of work, with consequent improvement in reliability.

In contrast with radio and television production, airborne electronic equipment is definitely a low-volume operation. For economic reasons, then, automatic production equipment must be sufficiently flexible to handle a wide variety of products with an absolute minimum of changeover time. One technique which appears to meet this requirement perfectly is etched wiring. Here a completely new wiring pattern is achieved simply by changing the master negative used for producing the desired pattern on the copper laminate. Dip soldering is likewise a logical operation, wherein all soldered joints are produced in one simple operation that gives a high degree of uniformity and reliability.

New Machines for Automatic Production

Automatic assembly, involving machines that insert parts in etched wiring panels in much the manner of a stapling machine, is already in use on high-volume commercial electronic products and is receiving considerable attention for military products as well. These new machines are appearing in various forms. Some take standard components directly, while others require that the components be belted together with tape, linked together by their leads to form chains, or preloaded in magazines.

In the field of automatic testing, curiously enough, military production is ahead of commercial. One large military manufacturer has for some time been using two highly flexible automatic test setups. One is for actually evaluating performance and printing

TO PROGRAM JOB FOR ASSEMBLY MACHINE, operator punches card to indicate assembly operations. Then card goes into reader box.

AUTOMATIC ASSEMBLY MACHINE takes orders from reader box. First Melpar model, shown here, with covers removed, assembles any of six different resistors at any of 12 possible positions on small etched wiring board. At the top are six component hoppers, one loaded with magazines. To the left of the hoppers is the wiring board dispenser. Below the hoppers is the manual control panel. Engineer at right holds completed assembly.

the results as a permanent record for use by the military inspector. This record pinpoints failure, and definitely expedites feedback of design defects. The recent development of digital voltmeters, which deliver pulses to operate a printer instead of moving a pointer over a scale, is largely responsible for the success of automatic testing. The machines are universal, and changeover of control settings for testing a new model takes only a few minutes.

Digital computer techniques are just beginning to be applied to the production of panels for electronic equipment. This involves adoption of integral dimensioning, in which 0.1 in. is emerging as the industry standard. It is envisioned that some day soon the engineering department will send punched paper tapes rather than drawings to the production department, for immediate use in controlling the automatic machine tools that punch and stamp the sheet material. This brings up the question of how the government can be supplied with the drawings it now requires; will there have to be an automatic drafting machine capable of producing drawings from punched tape?

Etched Wiring Wins Approval

Although very little information on the performance of etched wiring has come back from the field as yet, initial reports are encouraging. The usual open joints and corrosion problems occur, but these are expected and largely familiar. The combination of dust and high humidity on the insulating base for etched wiring appears to be the most important problem at the moment.

So far there seems to be no really adequate technique for coating an etched wiring board to minimize adverse effects of humidity, other than complete embedment. This requires considerable thickness of coating, adding to bulk and weight.

There has been surprisingly little indication of separation of the conductor from the base, even where adhesion is low. Conductors have broken because of warping but this is largely due to the action of heat on a base material that lacks dimensional stability.

There is no agreement on optimum temperature for dip soldering of etched wiring. Some solderers encounter trouble when soldering quickly at high temperature, and others have trouble while dipping longer at low temperatures. It must be recognized that phenolic material is just naturally not suited for dunking in solder under any conditions.

Although there is little if any printing of components directly on etched wiring panels, this technique has been used successfully for subassemblies of two or more components. It is expected that more and more components will be printed directly over circuit wiring as techniques are further perfected. The chief problem here is economic in that at present printed components require a ceramic base because of the need for firing. This ceramic is much more costly than a comparable plastic base.

(The report on which this article is based is available in full in multilith form together with reports of six other panel sessions of the 1955 SAE Production Forum held at the SAE Golden Anniversary Aeronautic Meeting, New York, April 21, 1955. This publication, SP-311, is available from SAE Special Publications Department. Price: \$1.50 to members, \$3.00 to nonmembers.)

Helicopter Horizon . . .

. . . appears rosy even though suitable transport helicopter has yet to be developed. Here are requirements for tomorrow as laid down by operators.

Based on paper by **Grahame H. Aldrich,** Air Transport Association of America

THE helicopter transport of tomorrow must be capable of carrying 35 to 50 passengers, plus baggage, plus 15% by weight of cargo—or a total design payload of 8000 to 11,500 lb. This is the size and capacity requirement specified by the Rotorcraft Committee of the Air Transport Association.

The Association's performance requirements are as follows:

1. The desired minimum cruising speed at 95% of maximum certified take-off weight, 1000-ft altitude, and standard NACA air is 130 knots.

2. The vertical rate of climb at sea level with maximum-except-take-off (METO) power, all engines operating, shall not be less than 600 fpm, without ground effect, at zero forward speed.

3. The hovering ceiling at METO power and 80% maximum certificated weight shall not be less than 8000 ft.

4. In case one engine becomes inoperative during the critical ascent period immediately after take-off, the rate of descent shall not exceed that which will seriously damage the aircraft.

5. Take-off performance with all engines operating should be obtained without the use of water under 100-F day conditions and without payload penalty.

6. Cruising speed shall be obtained at a power output low enough to be compatible with best possible operating reliability, and in no case shall cruising speed be greater than permitted by the maximum warranted cruise power of the engines as set by the manufacturer for commercial operation in transport helicopter service.

The range of this transport helicopter is to be 250 nautical miles with maximum payload plus 50

nautical miles alternate, and 20 min fuel reserve, against a 20-knot wind.

Although such requirements cannot be fully satisfied by any helicopter flying today, the manufacturers have come close to it with the S-56 Marine Assault helicopter and the YH-16 Transporter. There is no valid reason, technical or otherwise, to prevent the fulfillment of the ATA's requirements prior to 1958–1960. (Paper "Helicopter Horizons" was presented at SAE Golden Anniversary Aeronautic Meeting, New York, April 20, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Production and Material Control . . .

. . . calls for early decision on policy of making or buying—and for communications which are readily understood all along the line.

Based on panel secretary's report by **A. Farnette,** Ford Motor Co.

ONE very important decision which has to be made as part of production and material control is whether to make or to buy. Choice will depend heavily on such factors as overhead, absorption, outside supply availability, tooling, know-how, and quality. And where the guaranteed annual wage is a reality, that, too, will now influence the decision.

In some instances a decision will be made to make *and* buy. One aircraft engine plant, for example, makes approximately 50 units to establish sound basic quality and to develop new designs, while approximately 200 more units are subcontracted, at least in part. A producer of heavy-duty trucks, on the other hand, finds make-and-buy justified only in exceptional cases since about 50,000 parts are active and needed for fabrication in a highly integrated plant, but each part is wanted in relatively small volume.

One reason advanced for practicing make-and-buy is to utilize the engineering skills of both customer and supplier. The opportunities, however, are thought to be very limited except for high-volume items. Internal and external competition is generated, but competition is often keener between plants within the same company.

More clarity in specifications contained in purchase orders and requistions is sorely needed, and there could be improvement in legal conditions and packaging instructions, according to a steel mill representative. The greater use of coil steel rather than flat stock in connection with automatic machinery, for example, requires careful analysis of packaging specifications.

Material purchase can be influenced greatly by the type of manufacture. In the case of one large machine tool builder, operations are essentially those of a glorified jobbing foundry, involving problems of extreme seasonal demand and special machines. Here, because of the relatively small use for each possible specification of material, initial cost is sacrificed through general purpose purchasing. This results in having fewer sizes and specifications of bar stock to be procured and stored, but with more machining to fit the needs.

The flow system of planning and scheduling as employed by the automotive industry has two basic principles: (1) the assumption that all operations take place within a single plant regardless of ownership or location and that all forms of transportation such as boats, trucks, and railroads are in-plant conveyors (2) scheduling toward an accumulative total of products to be delivered, with periodic revision of instructions through superseding rather than amending paper. Under this system, inventories remain automatically at the desired level as long as the schedule is met. It is impossible, however, to forecast sudden changes in customer demand, strikes, fires, or similar troubles; hence improvisation and special expediting are still necessary, though held to a minimum.

In the last analysis production and material control is quantity control and since quantity control extends all the way from engineering specifications to delivery of the finished product, a closely coordinated line of communications and information is imperative. And whatever the type of communication, the best is probably the most readily understood.

This article is based on papers presented by a panel on "Production and Material Control" held as part of the SAE Golden Anniversary Production Meeting and Forum, Cincinnati, March 14, 1955.

Panel Leader:

F. C. McCoard, Ford Motor Co.

Panel Secretary:

A. Farnette, Ford Motor Co.

Panel Members:

J. E. Adams, White Motor Co.

A. H. Davis, R. K. LeBlond Machine Tool Co.

R. L. Grunewald, General Electric Co.

J. G. Wikoff, Armco Steel Corp.

E. A. Wright, North American Aviation, Inc.

The report on this panel is available in full in multilith form together with reports of the seven other panel sessions. This publication, SP-310 is available from SAE Special Publications Department. Price: $1.50 to members; $3.00 to nonmembers.

FATIGUE

. . . what we do and

IF a metallurgist tells you that fatigue is one of the most elusive and peculiar properties of metals, he has good reason for saying so. For instance:

1. It's been studied more extensively than any other strength-of-materials problem.

2. There isn't any precise correlation between fatigue and any of the other more readily determined properties.

3. Despite the many theories of fatigue, no one has yet fully explained its behavior.

4. Even laboratory fatigue tests don't give the complete picture of fatigue strength, and a factor of safety in design is a must.

Fatigue is defined by Webster as exhaustion of strength from labor, toil, or exertion of any kind. This is not an engineering definition. But it does indicate surprisingly well what happens when a structural part or a machine element fails from fatigue . . . namely the complete loss of strength as the result of a part being worked.

However, in the engineering sense, fatigue occurs only as a result of a special type of working, namely cyclic stressing. This cyclic stress may be applied very slowly, very rapidly, intermittently, or as a stress spectrum which is difficult to reproduce experimentally. These stresses may be caused by vibration, rotation, reciprocating motion of machine elements, alternate loading and unloading, thermal changes, by magneto-striction, or otherwise. Yet irrespective of the source of the cyclic stress, the phenomenon of the progressive fracture of metal by means of a crack which is propagated by repeated cycles of stress or strain is called fatigue. This is a discrete property of metals. Despite many attempts to prove otherwise, no one has been able to show that there is a precise correlation between fatigue and any other physical property.

1. How Fatigue Is Born

Fatigue failures are probably due primarily to progressive strain hardening, particularly in areas of high stress concentration. It can arise from material defects, design features, fabrication accidents or surface damage during use. In service an infinitely small amount of metal usually at or near the surface, is subjected to stresses above the fatigue strength of the metal and it fails. The volume of metal originally affected at the start of the fatigue failure is so small that, up to the present time, it has not been detectable by metallographic examination, x-ray, sonic or ultrasonic methods, loss in stiffness, dilatometry, treatment with magnetic particles, or by any other method until after irreparable damage has been done.

John A. Bennett, working at the National Bureau of Standards, developed a technique for detecting microscopically the early appearance of circumferential cracks in rotating beam specimens of steel. This technique indicated that the cracks could not be detected until their length was at least sufficiently long to cover 6 deg of arc, and that no significant changes in properties could be detected from load deflection tests on such specimens until the cracks were sufficiently long to cover approximately 60 deg of arc.

More recently Hunter and Fricke have used both the metallographic microscope and the electron microscope to determine the beginning of slip and the point of slip saturation on polished and etched

George R. Gohn, Bell Telephone Laboratories, Inc.

Excerpts from paper "Fatigue and Its Relation to the Mechanical and Metallurgical Properties of Metals," was presented at a meeting of the Shot Peening Division, of the SAE Iron and Steel Technical Committee, Hot Springs, Va., Sept. 28, 1954. This paper will be published in full in the 1956 SAE Transactions.

don't know about it

aluminum alloy fatigue specimens. Curves of these values plotted against the logarithm of cycle life are similar to the conventional S-N diagrams. There appears to be a definite correlation between the two, although the slip saturation curve tends to become asymptotic at a lower stress value than does the S-N curve.

Present test methods will not detect the failure of such an infinitesimal amount of metal as that which marks the beginning of fatigue. But it appears quite plausible to assume that the local failure at the submicroscopic level automatically increases the stress at that point. As the stress increases, the crack progresses until the section becomes too weak to withstand the load, when complete failure occurs.

The fatigue failure occurs with little or no deformation and seldom with any warning. The fracture itself is generally transcrystalline and may show two or more distinct zones . . . first a smooth area showing concentric or oyster-scale markings starting from a nucleus or stress raiser, and second a rougher area which frequently shows bright crystalline facets. There may also be present a discolored area where, under progressive fracture, the two surfaces have rubbed together. In the absence of corrosion, usually only one crack is observed; but where corrosion is a factor, many cracks are present which start from microscopic corrosion pits.

For most metals the frequency of the applied stress has but little effect on fatigue life in the absence of creep or corrosion. However, when one or both of these two factors are present, more cycles-to-failure are usually observed at faster testing speeds. That's because both corrosion and creep are time-dependent functions. So their effects are less pronounced at high testing speeds.

Since fatigue failure occurs as a result of extreme stress concentration, in this respect it resembles the failure of the notched impact bar. The only difference—but a most important one—is that of time.

In the fatigue test the specimen will hold together for a period of time after initiation of the first crack. This time interval may be relatively long or short in the case of fatigue, depending upon the properties of the material, the magnitude of the localized stress, and the frequency of the stress fluctuation. But in the impact test, failure takes place almost immediately because higher stresses are developed and the stress gradient is greater. Therefore, the crack is propagated faster than is the fatigue crack. In neither case is there any appreciable deformation nor any warning prior to failure.

2. Where Theory Falls Short

While the progressive strain hardening theory explains many of the phenomena observed in fatigue there are some which this theory does not explain. Kenyon observed that hard-drawn copper wire subjected to reversed bending stresses in rotating beam fatigue tests developed "islands" of soft copper at the periphery. Polakowski and Palchoudhuri reported a similar softening in cold-worked copper, Grade A (5% tin) bronze, 80-20 cupro-nickel, 2S and 3S aluminum alloys, and low carbon-titanium steel rods tested in alternate tension and compression.

These workers have suggested, therefore, that:

The progressive strain hardening theory applies only to soft or unworked metals. For the cold-worked metals, and by implication for the heat-treated metals, the formation of fatigue cracks is associated with the semi-plastic condition developed by the action of alternate strains.

Two recent symposiums have shown that fatigue

is a statistical phenomenon. At high stresses, in the finite life range, our own tests indicate that the number of cycles-to-failure has a log-normal distribution for any given stress. At lower stresses approaching or corresponding to the infinite life range, the distribution of the number of cycles-to-failure is no longer log-normal. But as yet, the probable nature of the distribution has not been established.

3. The Dislocation Theory

An explanation has been suggested for the mechanism of crack formation in fatigue based upon the recently developed concept of dislocations. Slip, which appears to precede the formation of a fatigue crack, is conceived to take place by the propagation of dislocations. This lattice defect in its simplest form consists of a missing half-plane of atoms in a part of a crystal. Thus, on one side of the slip plane the lattice is compressed so that (n + 1) atoms are opposite (n) atoms on the other side. This causes a distortion over a limited distance in the lattice, and the distortion can be moved along the slip plane, producing microscopic slip, with relative ease.

Orowan has shown how the piling up of dislocations against an obstruction—a precipitate particle, for instance—can lead to the formation of a crack perpendicular to the slip plane. The same author has explained how a grouping of dislocations can result in a stress pattern which will tend to open up a crack along the slip plane.

While dislocation theory has not been successful in providing a complete theory of fatigue failure, there is justification for applying it in a somewhat speculative form. Increasing evidence indicates that metals do deform by means of dislocations. Therefore, any theory of fatigue must take them into account. It is possible that only when the behavior of dislocations under cyclic stressing is understood will the theory of fatigue failure be in a satisfactory state.

4. No Kinship With Other Properties

Many mechanical properties, including the fatigue characteristics of metals, have been determined and are available to the designer. Some of these properties are constant, others vary from lot to lot with slight changes in composition, or with variations in heat-treatment, or in the amount of cold-working. Those which vary to any extent need to be determined for each lot of material. Other more constant properties need only be determined occasionally.

Unfortunately, those properties which are usually determined in inspection are of little use to designers. Determination of those properties required for sound design either involve precise laboratory equipment and techniques not generally available in the routine inspection laboratory, or take a long time for their evaluation. The properties normally used for inspection include hardness or tensile strength, elongation (or reduction in area) and yield strength.

Occasionally electrical resistivity (or conductivity) and notch impact strength are also acceptance requirements. Useful engineering properties which are usually determined in the research laboratory include the modulus of elasticity, creep strength and fatigue. Any appraisal of the adequacy of the material for use in engineering design should be made in terms of stress or strain and resistance to stress or deformation.

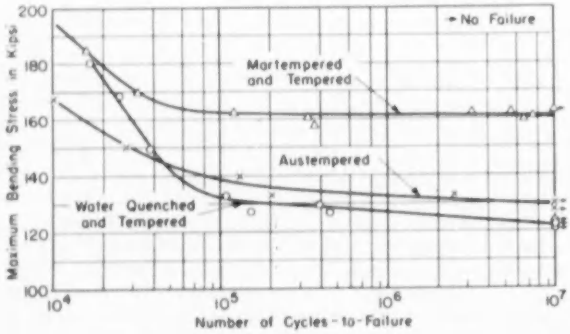


Fig. 1—Hardness is no index of fatigue, these S-N diagrams show. These curves are for three samples of SAE 1095 steel, all having hardness of RC 53. Yet the fatigue properties of each varied considerably.

Obviously **hardness,** the most commonly used acceptance test, does not meet this criterion. The hardness value, while perhaps conveying useful information about machinability and wear, is not itself directly applicable in design because, it like tensile strength, involves deformation far greater than that which can be permitted in service. Furthermore, hardness cannot be used as a ready means of evaluating fatigue strength. For instance, Fig. 1 shows that materials may have the same hardness, but widely different fatigue properties.

The tensile strength test, which is the basic acceptance test for practically all metals, is measured in terms of stress. But since it is a measure of failure, it is a value which seldom can be used in design. Complete rupture in engineering applications occurs only in protective devices such as shear pins, fuses, bursting discs, and such things as high explosive shells. Everywhere else the engineer designs for life, not failure.

Because the tensile test is so widely used in the inspection of materials, considerable effort has been made to correlate tensile strength with fatigue properties. In the case of low carbon steels the correlation is fair. As may be seen from a study of Fig. 2, Bullens has shown graphically that the endurance limits of polished specimens of carbon steel approximate 50% of the tensile strength up to values of about 180,000 psi, after which the ratio decreases rapidly. However, there is a wide scatter band on both sides of the average line, which makes it dangerous to use this relationship for intelligent design.

This tensile strength-fatigue relationship also

applies only to polished specimens which have had all of the grinding marks removed by a final longitudinal polishing operation. In the presence of notches, the fatigue strength may be much less. In the case of corrosion, a still greater loss in fatigue strength may be anticipated.

It has been shown that the bending fatigue strength of "as rolled" copper base alloys in the form of strip varies from as little as 14% to as high as 44% of the ultimate tensile strength for cold-rolled or cold-rolled and annealed materials. Whereas, the tensile strength of these materials could be more than doubled by cold-rolling the alloys 8 to 10 numbers hard (60.5–68.0% reduction), the fatigue properties were increased only 30 to 50% by such cold working.

This also is true, in general, for the aluminum alloys insofar as the effect of cold-working is concerned. But in the case of some of the heat-treated aluminum alloys, the improvement in fatigue properties resulting from heat-treatment more nearly approximates the gain in tensile strength.

Gohn and Arnold, in studies on beryllium copper, showed that by heat-treatment the tensile strength could be more than doubled, but the increase in bending fatigue strength was less than 20%. A similar lack of correlation between fatigue and tensile strength for machined and polished specimens of various non-ferrous materials is reported by Anderson, Swan and Palmer.

While it appears that the fatigue strength is somewhat related to the tensile strength of the material, **it would be dangerous from a design standpoint to assume that the fatigue strength can be estimated from the tensile test.**

Ductility values (elongation and reduction of area), frequently determined during the tensile test, cannot be used in design by the engineer to measure resistance to stress or strain. Nor can they be used directly by the manufacturer to evaluate cold-forming or the drawing properties of a material. According to Gillett, both are used for what they connote rather than what they state.

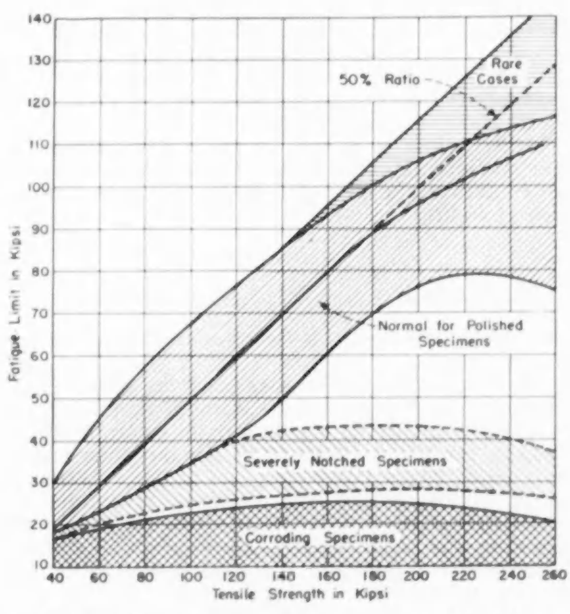


Fig. 2—Charted here is the behavior of wrought steel in fatigue as compared with its tensile strength.

While elongation or reduction of area cannot be used even indirectly to measure toughness, they do give some idea of the reserve of plasticity available for insurance against shattering fracture in an accident. Some ductility is beneficial in fatigue because, under conditions of repeated stress, local yielding at the base of notches permits a redistribution of stress which may prevent or delay fatigue failure. For example, a hardened steel used as a shaft or a bolting material may have a fatigue life no greater than that of a mild steel part because of its increased notch sensitivity.

In the case of low ductility materials such as the hardened steels, stress concentration at rough machined surfaces, at re-entrant angles, key ways, threads, and so forth, may start microscopic cracks which then develop further until failure results. But in the case of parts made from a more ductile material, the stress concentration may result only in local yielding and redistribution of stress. This is particularly evident in redundant, built-up structures such as welded, riveted, or bolted assemblies. **Here the more ductile materials may yield sufficiently to redistribute the stress in the entire structure, whereas high-strength materials of low ductility fail.**

This point should be kept in mind, particularly in aircraft design, where there is an increasing tendency to use light alloys of higher and higher tensile and yield properties gained at the expense of ductility.

In general, the primary criterion of load-carrying ability is **yield strength,** . . . the maximum stress that can be safely carried by the machine element without any permanent deformation. This property is frequently determined during inspection, except for the case of the mild steels and a few similar materials. But it is an arbitrary value based upon a given offset of the load-strain curve from the extension of the straight-line portion of the curve, or upon a specified total elongation under load. Little work has been done to establish the relationship between the yield strength and fatigue.

Templin in his recent Gillett Memorial Lecture showed that the yield strength of the aluminum alloys could be increased from 5000 psi for 2 S-O to more than 70,000 psi for 75 S-T as a result of alloying, heat-treatment, and cold-working. For the same alloys, the fatigue strength increased from 5000 psi to only 24,000 psi—a five-fold increase in fatigue compared to a 14-fold increase in yield strength.

About all that can be said is that under conditions of completely reversed stress, the fatigue limit is, with few exceptions, considerably less than the yield strength. For conditions of alternating or fluctuating stress superimposed upon a mean stress other than zero, the maximum stress amplitude in the cycle is limited by the yield strength. A similar lack of correlation exists between the elastic limit and the fatigue strength of metals.

Attention has been previously called to the fact that both fatigue and impact failures produce brittle, nonductile fractures. The similarity in the type of fracture might indicate some correlation between these two properties. It should be re-

membered that the notch brittleness of a material depends upon the geometry of the piece, the restraint this geometry of size and shape imposes upon plastic deformation, and upon the ability of the material to deform in a plastic fashion. Furthermore, **the fatigue strength generally increases with decreasing temperature, whereas the impact value of many ferrous alloys decreases under similar conditions.** It should also be observed that cast iron, which has very poor impact strength, not only has fatigue ratios approximating those of some of the steels of the highest impact strengths, but also exhibits less notch sensitivity than any other ferrous metal.

The significance of the impact test itself is still the subject of much controversy and it is doubtful if any correlation can be established between impact and fatigue strength.

The **modulus of elasticity** of the material, which measures the elastic deformation per unit stress, is seldom determined in routine inspection because it is essentially constant for a given class of material. It is not subject to control by the metallurgist or the metal manufacturer. Variations in composition, cold-working, or heat-treatment may cause slight variations in modulus. But in most cases the values are so nearly constant for any given class of material that previously determined values can be used by the designer to calculate the stiffness of his structure without the introduction of serious errors.

For example, the modulus of the carbon steels and most of the alloy steels is 29 or 30 million psi; for the aluminum alloys it is 10.3 million psi; for the magnesium alloys 6.5 million psi; for the brasses and bronzes about 15 million psi.

The modulus values are affected slightly by such things as changes in composition, heat-treatment, and cold-working. But the effect is small compared to the effect of these variables on the fatigue strength. On the other hand grain direction generally has a far greater effect on the modulus of elasticity than it does on fatigue strength. It has been shown that composition and heat-treatment markedly affect the fatigue properties of steels, but not the modulus, and that cold-working, composition and sometimes heat-treatment affect the fatigue strength of nonferrous alloys without affecting the modulus. **Hence it would appear that there can be no relation between the modulus of elasticity and the fatigue strength.**

Creep and fatigue have been related in some of the work done at the National Advisory Committee for Aeronautics by Machlin and others. However, the determination of the creep properties is a far more time consuming and critical operation than is the fatigue test. At present, creep properties are not sufficiently well established to permit their use as a basis upon which to estimate fatigue life.

Rather fruitless attempts have also been made to correlate **damping capacity** with fatigue. However, in the case of structures such as line wire, cable, and so forth, a high damping capacity may prevent the building up of dangerous stress amplitudes under external forces such as wind or vibration, which would themselves lead to premature failure of the structure.

5. What S-N Curves Tell Us

The comparison of fatigue strength with the more readily determined mechanical properties of metals indicates that fatigue may not be evaluated except by independent tests upon the materials themselves. The basic fatigue properties of any metal can be portrayed by a S-N diagram in which stress (S) is plotted against the logarithm of cycle life (N). If the S-N curve for a given material becomes asymptotic to the cycle axis, that material is said to have a fatigue limit.

For most materials this condition is never realized —at least not within a reasonable finite time. This limit is not reached in the case of aluminum and magnesium alloys, even after 500 million cycles. In the case of the copper-base alloys, the curve is essentially horizontal after 100 to 300 million cycles of repeated stress; although here again a true fatigue limit is seldom realized. In the case of the wrought steels, however, the fatigue limit is approached in conventional tests after about 12 to 20 million cycles. But in corrosive media even these steels show no fatigue limit.

Nevertheless, it is well to remember that, **while few materials have a fatigue limit, they all possess a definite fatigue strength corresponding to a specified cycle life.**

6. Some Factors That Affect Fatigue

Grain size, particularly the grain size in the ready-to-finish anneal, markedly affects fatigue. Higher values of fatigue are usually associated with small grain size material. The effect of grain size is just the opposite on creep. For the latter, best results are obtained with large grain size material. Hence in a structure such as a gas turbine or a jet engine where both creep and fatigue are factors, the optimum grain size must be a compromise.

Work-hardening processes such as cold rolling, cold drawing, and shot peening have all been used to increase the fatigue properties of metals. But too heavy reductions or excessive shot peening may exhaust the ductility of the metal to the point where cracks are formed or where a low stress may start a fatigue crack.

While most fatigue cracks start in areas where tensile stresses predominate, torsion fatigue cracks have been known to start in areas where shearing stresses predominate. H. F. Moore has shown some rail heads where failure appears to have started in areas where compressive stresses predominated. Nevertheless, it is generally concluded that **cold working—particularly cold-working which induces compressive stresses in areas subjected to fatigue stress—is beneficial.**

Since practically all fatigue failures originate at or near the surface, **surface finish** is bound to have

a marked effect on fatigue. The effect of surface finishes such as those used to prepare fatigue specimens for test is shown in Fig. 3. Surface treatments such as anodizing, pickling, and so forth, may also have an adverse effect on fatigue life. This is illustrated in Fig. 4, which shows the effect of various standard surface treatments on the fatigue life of some forged magnesium bars. Protective finishes, particularly those with high residual tensile stresses, may also affect fatigue life.

The **size of the fatigue specimen** also affects the fatigue life. H. F. Moore, in tests on a series of steels, showed that fatigue limits decreased 15 to 20% as the size of the test specimen increased from 1/8 to about 1 in. in diameter, but showed no further decrease up to 2-in. diameter sections. Much larger sections have been shown by Horger and others to have lower fatigue strengths. This may be attributed to variations in the steel throughout large sections.

The **type of machine** used to determine the fatigue properties may likewise affect the results. For example, direct axial stress machines may give results that are as much as 30% lower than the results obtained in rotating beam tests. The results of flexural tests are usually somewhat higher than those obtained in rotating beam tests. The low results on axial fatigue tests may be due to difficulties in avoiding some eccentricity in loading.

The **speed of test** has but little effect on the fatigue limit of most metals tested at room temperature. On the other hand, metals like lead, which are plastic at room temperature, are markedly affected by the speed of test. In such cases high testing speeds usually result in greatly increased cycle-life. At elevated temperatures, such as those in gas turbines, the fatigue properties of other metals are undoubtedly affected in a similar manner.

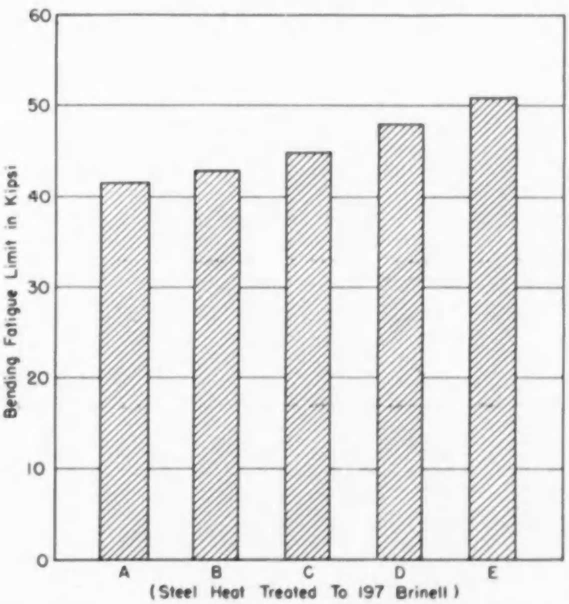


Fig. 3—Surface finish has an effect on fatigue. These rotating beam test data are on 0.49% carbon steel. Specimens were finished as follows: A—rough turned; B—smooth turned; C—ground; D—00 emery finish; and E—high polish, longitudinal.

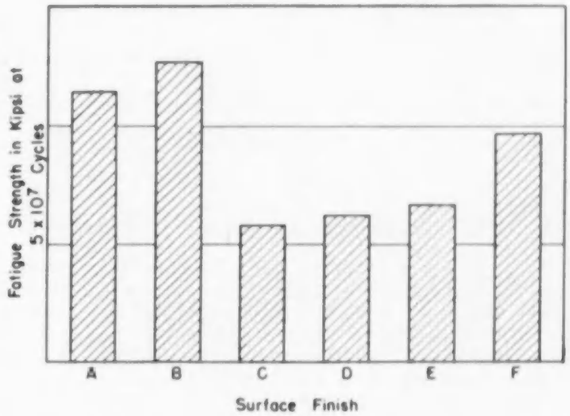


Fig. 4—Surface treatments also influence the fatigue strength of a material. For instance, in the case shown here, extruded magnesium bars had different fatigue strength with various finishes. "A" was smooth machined; "B" was polished; "C" was nitric acid etched; "D" was sulfuric acid etched; "E" received a caustic soda etching; and "F" was nitric acid etched and given a hot chromate treatment for 1/2 hr.

The **effect of corrosion** has been shown to be associated with the formation of corrosion pits. The result is similar to the notch sensitivity in uncorroded specimens. The effect of notches and notch sensitivity has been studied intensively and stress concentration nomograms have been developed by H. Neuber for evaluating the effect of notches on round shafts—either solid or hollow—when used in torsion, in bending, or in tension.

The **effect of mean stress** on the stress range is of primary importance. In most cases the range of stress decreases with increasing mean stress. But as Gohn and Ellis have demonstrated, this is not true of bending tests on nonferrous sheet metals, nor is it true for torsional tests on ferrous metals. Since stress conditions may vary, it is essential that the fatigue properties be established for various mean stresses. This means the determination of fatigue diagrams.

The influence of stress cycles of different magnitudes applied in succession and after rest periods is important, but few studies of this type have been made. Yet this is typical of service conditions.

To sum up, our present knowledge of fatigue is largely determined from laboratory tests on highly polished specimens tested either in bending (rotation or flexure), tension-compression, or torsion under conditions of constant deflection or constant load until failure occurs. The standard specimen is an ideal one, however, and seldom simulates service conditions.

The surface of the test specimen is usually very carefully ground or longitudinally polished to eliminate stress raisers. Since the latter are nearly always present in machine parts due to machining, grinding, or other fabricating operations, or from the use of threads, collects, or splines, and so forth, we must allow for a factor of ignorance when the results of laboratory fatigue tests are applied to our designs. Furthermore, allowance must be made for differences in size between test specimen and machine elements which introduces still another variable.

So even when used as an evaluation test, **a fatigue**

test yields an ideal fatigue strength which cannot be used in design without a factor of safety, the magnitude of which can only be determined from valid life test data.

While it is possible to increase the fatigue strength of a metal by heat-treatment or cold-working, it should be remembered that the harder materials are usually more notch sensitive. So under corrosive conditions, they may be no better than a softer, more ductile material.

7. Shot Peening . . . How It Fights Fatigue

One of the most important steps in reducing the number of fatigue failures in moving parts would be to eliminate stress raisers. That is true, and yet Zimmerli, Moore, and Almen have all shown that by roughening a surface by shot peening, the fatigue life of a part may be greatly increased.

The effectiveness of shot peening in improving the fatigue resistance of manufactured parts is summarized by the ASM Committee on Shot Peening in a recent Supplement to the Metals Handbook. This article, which was prepared as an aid in selecting the most suitable process, evaluates shot peening, surface rolling, hand and mechanical peening, and prestressing. The data presented show that the fatigue strength of phosphor bronze sheet can be increased from 31,600 psi at 10^7 cycles of reversed stress to 47,000 psi by shot peening, a gain of 52%. Our own data, which are presented in Fig. 5 for two tempers of Grade A (5% tin) phosphor bronze, show that under certain circumstances even greater resistance can be imparted by shot peening.

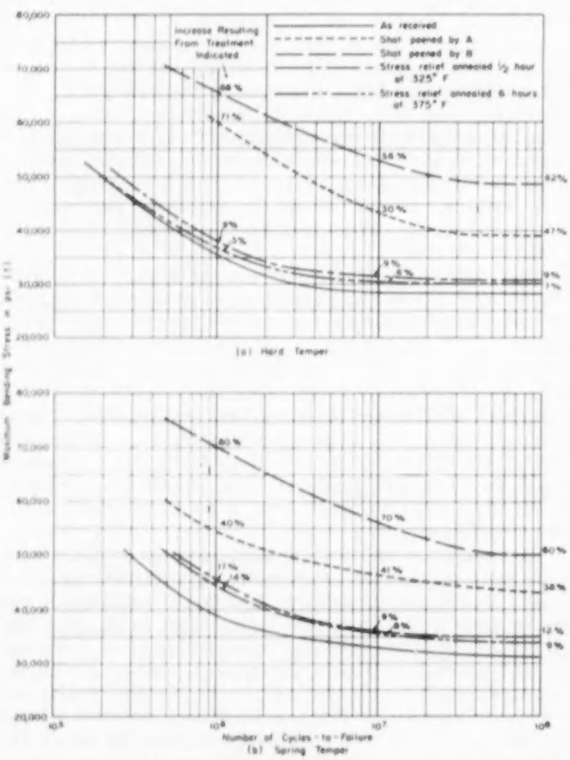


Fig. 5—Both shot peening and stress relief annealing improved the bending fatigue properties of phosphor bronze strip. The shot peening added desirable surface stresses and the annealing eliminated undesirable residual stresses.

However, the commercial application of such shot peening to thin sheet metal springs of low section modulus such as those used in communication equipment, is rather difficult. That's because of the close engineering requirements which frequently necessitate spring adjustments after assembly or during service. Such adjustments materially reduce the effectiveness of the shot peening.

The data presented in Fig. 5 are significant not only because they show that the fatigue resistance can be increased by *adding* surface stresses of the proper kind, but also because the data show that *elimination* of residual stresses by stress relief annealing will also be beneficial.

In the examples shown, the gain in fatigue resistance resulting from the elimination or reduction of residual stress by low temperature annealing is not as great as that resulting from the introduction of added compressive stresses by shot peening. This apparent anomaly on the effect of residual stresses clearly shows **the danger inherent in attempting to draw general conclusions from any given set of fatigue data.**

Vitovec offers a possible explanation of the beneficial effect of shot peening as well as other forms of cold working. According to his theory, the deformation of a polycrystalline material inhomogeneous, due not only to the statistical orientation of the grains, but also to the influence of the free surface. Because of the statistically crystallographic orientation of the grains, the slip planes do not coincide.

Stressing the material causes lattice distortions and a mutual hindering of slip of the grains. Any interior grain is strengthened by the hindering effect of the surrounding grain. For an interior grain, Vitovec shows mathematically that the total hindering effect is theoretically twice the relative hindering effect on a surface grain.

From this it follows that, if fatigue is caused by slip, then fatigue in a homogeneous material is a surface phenomenon and the fatigue of metals must be intimately related to the surface effect on plastic deformation. Therefore, the fatigue strength of metals can be increased by increasing the yield strength of the layer which is influenced by the free surface. Various processes, such as the use of electroplated finishes having residual compressive rather than tensile stresses, cold working (shot peening), aging, case hardening, carburizing, nitriding, and precipitation hardening, will increase the yield strength in this surface layer. This increase in yield strength will result in an increase in fatigue according to Vitovec. But the increase is only effective up to the fatigue strength of the material inside the specimen.

Extreme Temperatures Force Search For . . .

New Hydraulic Fluids and Seals

ONE of the problems facing the aircraft industry today is the need for hydraulic fluids and seals that can operate efficiently under the extreme temperatures encountered in supersonic, high-altitude flight. The SAE Committee A-6 on Aircraft Hydraulic and Pneumatic Equipment has been working closely with hydraulic engineers of industry and government to suggest improvements in design and avenues of development.

Currently aircraft hydraulic systems may be classified as one of three types:

Type I	– 65 to 160 F
II	– 65 to 275 F
III	– 65 to 400 F

In these systems the fluid temperatures in any part of the system should not exceed 160, 275, and 400 F, respectively. Generally speaking, most of the problems in the first two classes have been solved and aircraft hydraulic systems are now operating satisfactorily throughout those temperature ranges. There is still much to be done, however, in the third temperature range, and already hydraulic designers are worried about temperatures reaching 700 F.

Thanks to Our Sources . . .

THIS article is based on the minutes of the combined meeting of SAE Committee A-6 (Aircraft Hydraulic and Pneumatic Equipment) with hydraulic engineers of industry and government at Montreal, Canada, April 27–29, 1955; and a paper "New Gland Designs for High Temperature Aircraft Hydraulic and Pneumatic Seals," presented at the meeting by T. J. McCuistion, Parker Appliance Co. B. R. Teree, The Weatherhead Co., is chairman and F. H. Pollard, Republic Aviation Corp., is vice-chairman of this committee.

Hydraulic Fluid Problems

For the past four years, the United States Air Force hydraulic specialists at Wright Air Development Center have been working to develop a hydraulic fluid that remains stable through a wide temperature range (– 65 to + 400 F), and which will not attack the packing at high temperatures.

Of the various classes of compounds studied the organo-silicon fluids had the best over-all properties and looked most promising to be formulated into a good – 65 to 400 F hydraulic fluid. In this field, organo-silicones, silicates, and silanes were investigated, and the ortho-silicate esters provided the first experimental high temperature hydraulic fluid (MLO-8200). The fluid consists of a dimer silicate, hexa (2-ethyl butoxy) disiloxane as a base compound, a polymeric silicone as a viscosity index improver, and several additives employed as anti-oxidation and anti-corrosion inhibitors. MLO-8200's oxidation stability was considered adequate. In fact, most silicate-ester base fluids have good over-all properties up to 275 F, even though they are not very compatible with seals made of neoprene elastomer.

Industry, too, has been experimenting with silicate-ester base fluids. Unfortunately the ortho-silicate esters have poor hydrolytic stability and special care must be taken to prevent water contamination. At 400 F, less than 1% water in the system is enough to promote hydrolysis.

Work is continuing to develop a good high-tem-

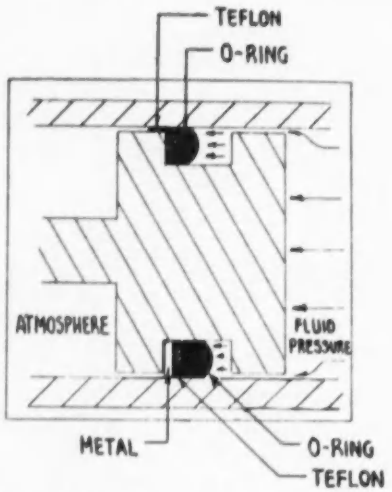


Fig. 1—Metal back-up rings have been used behind teflon o-rings to prevent extrusion at temperatures above 350 F.

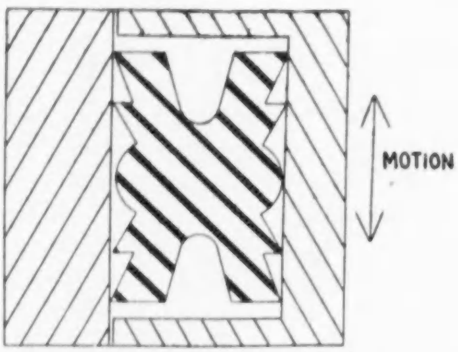


Fig. 2—The v-o-ring combines the advantages of the v-ring and the o-ring. When seal is compressed only the small points contact the metal surfaces.

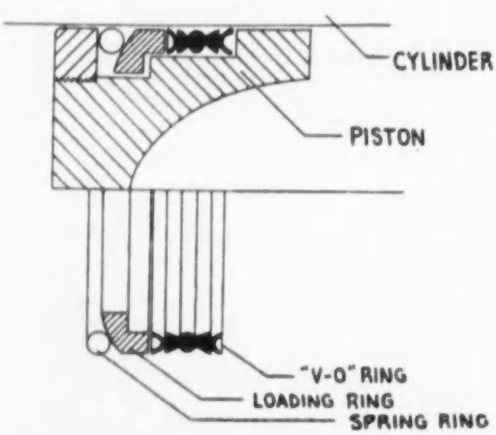


Fig. 3—A spring loaded v-o-ring may be used to seal moving glands at temperatures up to about 500 F.

perature hydraulic fluid and some engineers feel that if seals made of silicone elastomer instead of neoprene rubber seals, are used the problem can be solved. Currently, at temperatures up to 275 F, a standard petroleum-base fluid (MIL-O-5606) can be used. For temperatures up to 400 F, Monsanto OS-45, Hollingshead 720173, and WADC 85/15 (85% MLO 8200 with a high rubber swell additive) can be used. There are storage and handling problems inherent in these fluids which WADC is trying to solve.

Plans at WADC include blending the basic disiloxane used in the MLO 8200 with a different type ester that will provide more rubber swell with a smaller concentration of ester, and at the same time improve the low-temperature properties. Different inhibitors, which are more responsive to both silicate and carboxylic esters inhibition, will probably be added to the fluid. This should improve the hydrolytic stability.

Another project is to develop a fluid that is compatible with silicone elastomers. The silicone elastomer alleviates the high compression set shown by neoprene elastomers at 400 F. At the same time silicone rubber, which inherently possesses a wide temperature range, wouldn't require a plasticizer to prevent brittleness at very low (−65 F) temperatures. Preliminary results indicate that tetra-substituted silanes are possibilities.

It is particularly desirable that one, or at most two, hydraulic fluids be developed and adopted as standard. Then manufacturers of seals and other components of hydraulic systems can concentrate on developing materials and designs for the standard fluids instead of trying to accommodate many different fluids.

Seal Designs and Materials

Equally important as finding a suitable hydraulic fluid is the search for seals and packings that operate efficiently over ever widening temperature ranges. Seal materials must resist attack by hydraulic fluids; they must have proper flowability, and resist wear when used in moving glands.

High temperatures cause most rubber-like materials—even teflon—to soften and extrude. And very low temperatures make the materials brittle and rob them of sealing properties. Therefore, a combination seal is usually necessary, consisting of a seal to do the sealing and a back-up to resist extrusion. Usually a soft rubber seal material (less than 70 durometer) is desirable for the high-temperature sealing job. Even though it has less over-all resistance, it usually lasts longer at high temperatures and is more flexible at low temperatures.

Above 350 F, metal back-up rings have been employed behind teflon o-rings, as shown in Fig. 1. The teflon is too soft to prevent extrusion at these temperatures. A back-up ring resists extrusion in a high-temperature, high-pressure gland by plugging the clearance gap. It must be non-abrasive, adjustable so as to fit in slightly different glands, and be compact and easy to assemble.

Actually there are several high-temperature materials which in the future may eliminate the need for metal back-up rings. Many different compounds are being tested. North American Aviation, Inc., which has been conducting tests up to 400 F,

believes that the future will require seals not containing elastomers. Metal seals look promising; however, it is difficult to make a long life, completely leak-proof metal seal for a moving gland.

Republic Aviation Corp. has found that standard AN6290 boss seals are satisfactory at 400 F; however after 50 hours the rubber becomes very hard. Apparently it welds to the fitting. Other types of materials, such as asbestos-graphite, show promise at extremely high temperature.

But development of new materials alone is not enough. Glands and seals must be designed to make the most of existing materials and to take advantage of new compounds as they are discovered. Present materials can be made to function satisfactorily in reciprocating fluid glands from − 65 to 400 F or perhaps 450 F. It may be possible to use standard o-ring seals of materials like Kel-F in static applications. But new and probably more complicated seal and gland designs are believed necessary for moving glands.

The Rubber Division of the Parker Appliance Co. has been experimenting with some unorthodox seal designs. Fig. 2 shows what they call a v-o-ring, which combines the advantages of the v-ring with the o-ring. When the seal is compressed, only the small points will contact the metal surfaces. Twisting and resulting spiral failure isn't a problem because the seal length is approximately twice the width. It has low friction due to the small projected area of the seal (to the fluid) and compression requires less force. Pressure can deflect the end V-lips at very low pressures, fluid can be trapped in the V's to help lubricate an atmosphere-exposed surface on the return stroke. There are glands in present use that may find v-o-rings useful.

The best surface finish for an o-ring is a controversial subject. General practice is to make them perfectly smooth. However, there is one theory that a controlled rough-surface finished seal will enter into the micro-fine grooves of the metal—much like a broom's bristles enter into the cracks of a rough floor—thereby making a better seal. Theoretically, they are easier to seat because the hills and valleys of the rubber and metal fit together like threads of a bolt and nut. They seal better at low and high temperatures, they permit a harder seal material to be used, and they reduce friction slightly.

To seal moving glands at temperatures between 350 and 500 F, it may be necessary to adopt a more cumbersome, complicated spring-loaded seal, as shown in Fig. 3. The design should provide some lubrication for the seal regardless of whether fluid or gas is sealed in the other part of the assembly. Trapped and pressurized expandable gas keeps the seals spring-loaded. Positive sealing should be provided because the relatively large liquid molecules are easier to seal than gas molecules. Rod-type seals can be kept filled and pressurized from a central source instead of using a gas or pressurizing piston and spring as usually required in a moving piston gland.

Another type is the labyrinth seal. Some leakage must be tolerated with this type seal, but it is possible to use it up to 1000 F. Air which is blown into the center of the labyrinth seal, prevents the engine oil from escaping around a high-speed shaft.

At the top of Fig. 4 is a combination of several types of seals. Spring or pressure loaded (or both) metal foil offers the best material design possibility. It can be spirally wrapped around a mandrel then compressed into a small ring (while still on the mandrel) with many tiny loops at the top and bottom. Foil rings formed this way are somewhat flexible, and tiny loops appear to present a series of sharp sealing rings and/or notches to effect a labyrinth seal.

Assuming some leakage will occur, this type of gland is sealed by:

1. Blowing a gas through the two seals of the gland; thus preventing the fluid from entering.

2. Vacuum sucking the leakage from between the seal elements.

3. Using a fine powder or other similar flowable substance as a seal between the two primary seals.

4. Allowing controlled leakage of a nonflammable, noncharring substance.

Fig. 5 shows a cross-section of a design which uses the coefficient of expansion of the rubber (approximately 10 times that of steel) to advantage. As the o-ring shrinks, it should tend to wedge itself between the two surfaces and make a better (low pressure) seal. The principle has not yet been proved, but has possibilities.

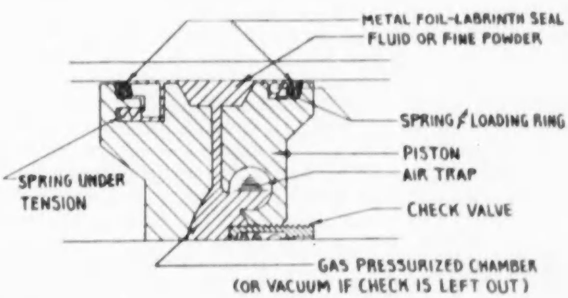


Fig. 4—At bottom, a labyrinth seal is shown. Engine oil can be prevented from leaking around a high speed shaft by blowing air into the center of a labyrinth seal. At the top is a combination of several types of seals, spring or pressure loaded, or both.

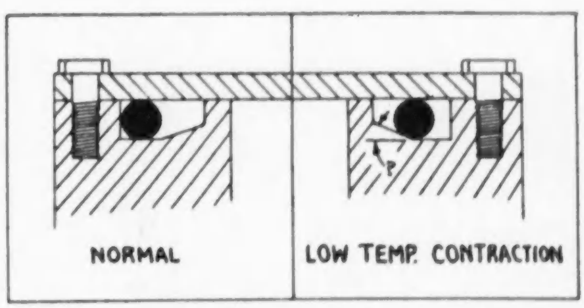


Fig. 5—Possible design: as the o-ring shrinks it tends to wedge itself between the two surfaces and make a better seal.

How Detergents Attack

Four Detergents, Including New Ashless Polymeric Type, Are Evaluated

DETERGENTS vary widely in their ability to keep an engine clean from sludge under low temperature conditions. Those which are effective at high engine temperatures are not necessarily effective at low engine temperatures. Bench tests show that many detergents lose their effectiveness in systems containing small amounts of water. And the type and amount of detergent in the engine oil will influence the benzene insolubles and the insoluble resin content of the oil.

The action of detergents

Materials which function as detergents in lubricating oil must possess at least one polar and one non-polar group. The non-polar group provides oil solubility for the molecule, while the polar group is of such a nature that it is attracted to the particles of sludge. These detergent molecules tend to cluster together to form micelles containing many molecules. Fig. 1 shows electron photomicrographs of two samples of new oil. One sample is a carefully filtered neutral, and the other sample is the same filtered base oil to which a barium petroleum sulfonate has been added. The micelles of the detergent are clearly visible as black specks in the additive-containing sample. They are 2 to 4 millimicrons in diameter and 5 to 20 millimicrons in length.

One way in which a detergent functions is by keeping sludge particles in suspension. The detergent molecules or micelles surround the sludge particles which develop in the oil. This coating of detergent prevents the particles from agglomerating. During the service life of the oil, the number of sludge particles gradually increases while the number of unattached detergent molecules or free-floating micelles decreases.

Fig. 2 shows photomicrographs of oil samples withdrawn from a passenger car engine operated in the laboratory on an oil containing ½% active ingredient of a barium petroleum sulfonate. The first picture shows the oil before it was placed in the engine, while the others show samples of the oil after 5, 15, and 40 hr of operation. It can be seen that as small particles of sludge are formed the micelles of detergent cluster around each particle and tend to keep them dispersed. As the oil continues in use, the micelles are gradually depleted and agglomeration of particles proceeds.

Some detergents are capable of keeping the size of sludge particles so small that, for all practical purposes, it can be said that the particles are in solution. This solubilization of sludge is the most effective way in which a detergent can function. Engine sludge in suspension in the lubricating oil does not tend to deposit on engine parts until the concentration becomes excessive. The deposition tendency of this sludge is reduced if the sludge is solubilized by the detergent.

Many laboratory engine tests and also road tests

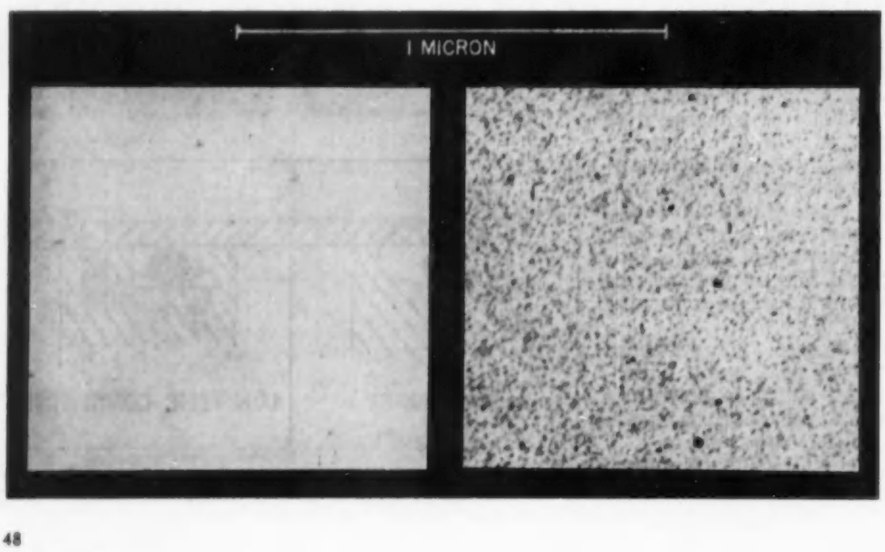


Fig. 1—Electron photomicrographs of oil without additive, on left, and with barium petroleum sulfonate, on right. Micelle formation is visible in the oil containing the detergent.

Sludge—

R. L. Willis and E. C. Ballard,

E. I. du Pont de Nemours & Co., Inc.

Based on paper "The Control of Low Temperature Sludge in Passenger Car Engines" presented at SAE Golden Anniversary Annual Meeting, Detroit, Jan. 11, 1955. This paper will be published in full in 1955 SAE Transactions.

have indicated that not all detergents perform adequately under low or moderate temperature operating conditions. The foregoing concepts of detergency are inadequate to explain these differences in performance. For this reason bench and engine tests were conducted with representative examples from four different classes of detergents having known differences in low temparature performance. The detergents studied were a basic barium phenate, a basic barium petroleum sulfonate, a metal-containing experimental detergent, and a nitrogen-containing polymeric detergent. This last detergent is a copolymer of lauryl methacrylate and diethylaminoethyl methacrylate having the structure shown in Fig. 3. It will be referred to as detergent C. The sulfonate and phenate selected for this work had the best low temperature laboratory engine performance of the five or six of each class investigated.

Bench tests

Bench tests were conducted to determine the ability of the four detergents to suspend sludge. Detergent A showed some suspending ability. Detergents B and D failed to keep sludge in suspension. Detergent C gave complete solubilization of the sludge.

In addition to the organic sludge which must be kept dissolved or in suspension, carbon from partially burned fuel must be prevented from depositing in the lubrication system. A test was carried out to determine the ability of the four detergents, at a concentration of 0.5% (by weight) active ingredient, to suspend a finely divided carbon in kerosene under both wet and dry conditions. The performance of detergents in the presence of water is considered to be important because water is present in crankcase oils during low temperature engine operation. This water enters the crankcase either as condensate from blowby gas during engine operation or as condensate which forms as a result of engine breathing during periods in which the engine is not in operation.

As shown in Fig. 4, detergents B and C maintained complete suspension of the carbon for more than six days, whereas detergents A and D lost their suspending power in less than one day. The suspending ability of detergent B was lost when 1% (by volume) water was present in the kerosene. Under these

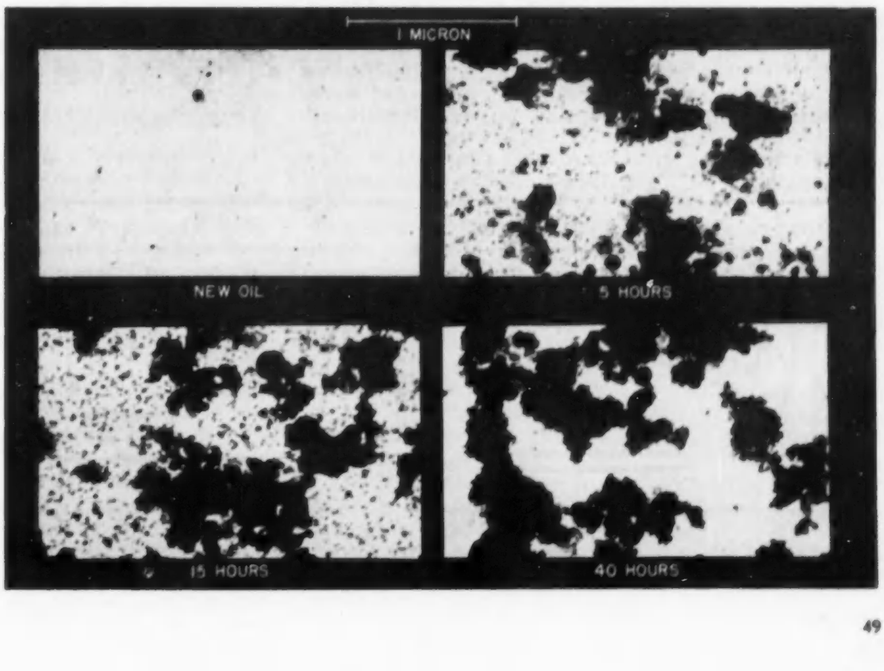


Fig. 2—Electron photomicrographs illustrating barium petroleum sulfonate micelle depletion in oil samples taken from a multicylinder engine test. As small particles of sludge are formed the micelles of detergent cluster around each particle and tend to keep them dispersed.

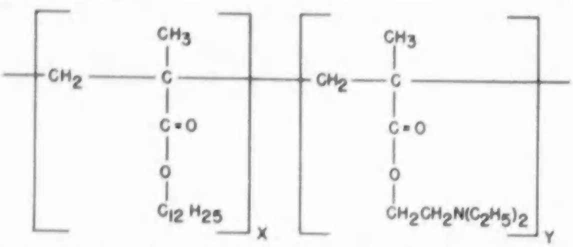


Fig. 3—Chemical structure of detergent C (lauryl methacrylate/diethylaminoethyl methacrylate).

conditions detergent C retains its effectiveness. These results are shown in Fig. 4.

Blotter tests

Blotter or filter paper spot tests were made to evaluate the residual detergency of the oils after being used. Detergents A, B, and C showed good detergency. D gave only marginal performance. The addition of ½% water neutralized the detergents A and D and decreased the efficiency of detergent B. Water had no effect on the dispersancy of detergent C.

Oxidation tests

The ability of the detergents to prevent the agglomeration and precipitation of sludge formed during the oxidation of lubricating oils was determined. Water increases the rate at which sludge develops. But detergent C minimizes sludge formation, even with water present.

Engine tests

These detergents were tested for sludge development in laboratory engines. At equal concentrations the different types of detergents vary widely in their effect on piston varnish and over-all engine score. Detergent C, as might be predicted from the previously discussed bench test results, was very effective in minimizing sludge and varnish formation. Although the oil containing detergent A actually gave a lower piston score than the base oil, it did show some improvement in over-all engine cleanliness. Detergents B and D had little effect on the oil performance under these conditions.

The tendency of the sludge to settle out of the used oils was studied visually following vigorous shaking of the samples. These settling tests were repeated several months later on the same samples with the same results, indicating that once the oils were removed from the engine their detergent qualities were quite stable. The settling rates obtained indicate that detergent A loses its effectiveness after six to eight hours of engine operation, detergent B at approximately fifteen hours, and detergent D at four hours. Detergent C effectively suspends sludge for the complete 40-hr test.

Growth or agglomeration of sludge particles normally occurs only during periods of engine operation as the detergent is gradually depleted. Agglomeration can also occur in the presence of some detergents during periods of non-operation, if the water content of the oil is increased by condensation or coolant leakage into the crankcase.

Sludge analyses of used oils from engine tests show that oils containing detergent C contain less oil-insoluble material than any of the other oils. This suggests that the type of detergent influences the benzene insolubles and detergent resin content of the oil. Detergents of the type D structure appear to increase the insoluble resins in the used oil, whereas detergents similar to A and B have little effect on these resins. Detergent C markedly decreases the insoluble resin content of the used oil. Besides detergent C's ability to solubilize sludge, it is possible that it also decreases the rate of the condensation and polymerization reactions which transform low molecular weight fuel and oil oxidation products into varnish and sludge. The low level of benzene insoluble material encountered in the engine oils with low insoluble resins suggests a relationship between the two materials. Indeed, it seems quite likely that the same reactions that lead to insoluble resins can contine and convert these resins into higher molecular weight, benzene insoluble materials.

Field performance of detergents

The results of the bench tests and laboratory engine tests have been substantiated by several road tests.

(Paper on which this abridgment is based is avail-

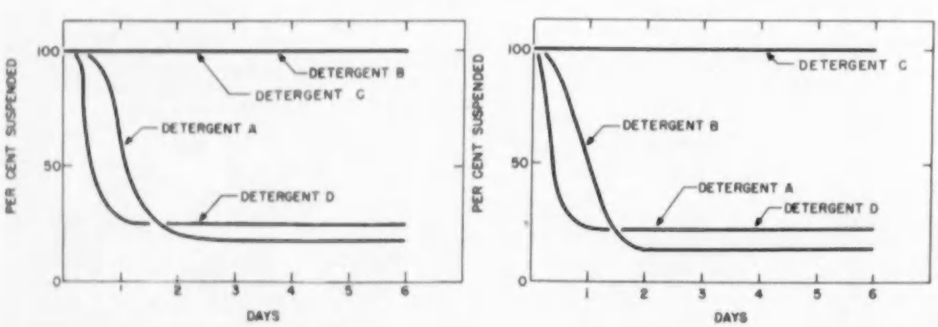


Fig. 4—The effect of water on detergent performance is shown here. At left, carbon is suspended in dry kerosene. When suspended in wet kerosene, at right, detergent B lost its suspending ability and detergent C retained its effectiveness.

able in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Based on Discussion . . .

J. G. Moxey, Jr.,
Sun Oil Co.

There are many performance features that these polymer detergents make no attempt to provide. But apparently they have made a real advance in providing engine cleanliness under the temperature conditions that we have to contend with in passenger car operation. Our tests with these additives have given pretty much the same results as described by the authors.

We should include in our additive development programs the search for materials aimed specifically at typical stop-and-go low temperature passenger car operation.

R. I. Potter,
Standard Oil Co. (Ohio)

The problem of low temperature sludge formation in V-8 engines is becoming more acute as engine horsepowers go up and the distance between stoplights becomes less. We conducted tests with an ashless detergent similar to the authors' detergent C. The ashless detergent was capable of suspending three to seven times as many oil contaminants as a metallic detergent, probably due to the fine dispersion of the solids. Further tests indicate that the amount of settling appears to vary proportionately with the detergent dispersant quality of the oil. Oil treated with the ashless type detergent shows very little separation.

We agree with the authors that the use of an electron microscope to determine the effectiveness of detergents to prevent sludge particle size growth in oil is a much more effective and novel method than waiting for used oil samples to settle.

Synthetic Lubricants . . .

. . . show satisfactory or almost-satisfactory performance in present gas turbines. But future will demand oils with higher gear loading capacities and better thermal stability.

Based on paper by **T. F. Davidson, T. P. Cooley,** and **J. H. May,** Wright Air Development Center

THE Air Force has been using synthetic gas turbine lubricants in quantities for the last four years. Here, specifically and briefly, is its experience:

Bulk oil temperature: The subsonic installations are generally satisfactory. The supersonic applications are definitely marginal. They can even be considered unsatisfactory if one considers the amount of supplementary air cooling needed in certain applications which could be eliminated with a more stable oil. Production turboprops are satisfactory at present.

"Hot spot coking": Experience puts this in a somewhat worse category than bulk oil temperature. In some applications there has been much trouble with coke. This necessitates short lubricant drain periods and frequent cleaning of oil filters and certain engine compartments.

Oil consumption: This is not a problem now, but excess consumption may be one in the future.

Film strength: Here we are in definite trouble. Some turboprop engines now have operating limits which could be raised with an oil of greater film strength. Main engine bearing performance is satisfactory in general.

Elastomers: With some of our engines this problem has become the limiting factor in increasing oil temperatures.

Low temperature properties: The lubricant tends to become marginal in some installations.

Foaming: No foam problem has been met.

Corrosion: No problems have been met.

Our experience with synthetic lubricants ranges from satisfactory to marginal. Much of the good experience can be attributed to very careful design of the lubricating system to eliminate non-compatible materials, provide supplementary cooling, and adequate shielding from hot sections of the engine. Such design consideration on the part of the engine and airframe builders has been extremely important and will become increasingly more so in the future. Experience thus tells us we could use better lubricants now and that we'll have to have them in the future.

Experience with materials in contact with synthetic oils has been varied and very frustrating at times. Our present MIL-L-7808 oils, for example, are not too compatible with elastomers, particularly at elevated temperatures. For this reason we are bulk-oil-temperature-limited in some applications, that is, we cannot operate above certain temperatures because of the oil-elastomer compatibility problem. The situation is aggravated by the fact that a number of very promising and economical base stock materials cannot even be considered for use in MIL-L-7808 type oils because of their effect on elastomers, although they would be satisfactory in all other properties.

The compatibility of materials with synthetic oils was primarily a trial and error experience. We know now of a large number of materials incompatible with synthetic oils and a select group which is reasonably compatible. The most comprehensive survey cataloging this experience, as far as we know, was made by the Aeronautical Standards Group, Department of Defense, in the middle of 1954. (Paper "Air Force Experience with Synthetic Gas Turbine Lubricants" was presented at SAE Golden Anniversary Aeronautic Meeting, New York, April 20, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

DISCUSSION ON NEXT PAGE

Based on discussion . . .

Max C. Hardin, Allison Division, General Motors Corp.

The problem of correcting turbine engine lubrication cannot be placed at the door of the lubricant producer alone. Undoubtedly some of the unsatisfactory experience reported has been or is in process of correction by mechanical design improvement. Experience emphasizes that the MIL-L-7808 material is marginal in several critical properties and that better lubricants will be required.

One approach worthy of consideration would be to separate the diverging lubricant requirements of turboprop and turbojet engines. The turboprop requires better load carrying ability and less emphasis on improved high temperature stability. The turbojet requires much better high temperature properties, while some decrease in film strength probably could be tolerated.

We need basic information concerning the effect of viscosity and lubricant composition upon the pitting fatigue characteristics of heavily loaded gears and bearings. Pertinent information is very limited, particularly within the viscosity range of these materials.

J. R. Morris, The Texas Co.

One of the most urgent needs is the development of new tests which will predict more adequately the performance of synthetic oils in actual service. As temperatures and demands on the lubricant increase in severity, the importance of such bench tests will loom larger. They will be of value only to the extent that they simulate actual engine performance.

It is our experience that one cannot predict the tendency of a lubricant to deposit sludge in ball bearings under high temperature, high load conditions on the basis of panel coking tests or similar thermal stability tests. A high temperature bearing test should be flexible enough to permit the testing of many materials under widely varying conditions.

CRC Reports

Octane Number Requirements Of Cars with New Design Engines and Transmissions

THE third year of a four-year cooperative program for investigating the octane number requirements of passenger cars has been completed and the results published in the CRC report 280, "Octane Number Requirement Survey, 1953."

The 1953 program was devoted to a study of the octane number requirements of cars with new design engines and transmissions. Twenty-four laboratories obtained data on a total of 235 cars, representing nine 1953 models.

Here in brief are the findings:

1—The Research method octane numbers of the 1953 reference fuels which satisfy 10, 50, and 90% of each of the nine models are given in Fig. 1. The models are listed in order of decreasing requirements at the 90% satisfied points, using full-boiling range gasolines. Requirements with full-boiling range gasolines were from 0 to 3 octane numbers higher than the primary reference fuel requirements at 10 and 50% cars satisfied and 1 to 4 octane numbers higher at 90% cars satisfied.

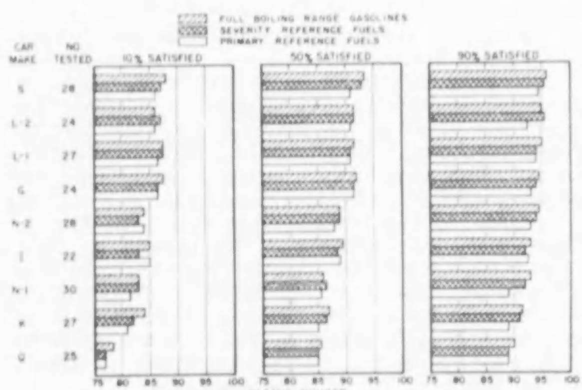


Fig. 1—Octane number requirement at 10%, 50%, and 90% cars satisfied.

2—The Research method ratings of full-boiling range gasolines required to satisfy 50% of the cars varied from 86 to 94 octane numbers for the nine models tested. The Research method rating required to satisfy 90% of the cars of a given model ranged from 90 to 96 octane number.

3—In general the severity reference fuel requirements were within the range covered by the primary and full-boiling range fuels.

4.—Six of the 1953 models tested were of the same make as those tested in previous surveys. One model, with no significant design changes between 1952 and 1953, showed substantially no change in requirement in the 50 to 90% cars satisfied range. The other five models which did have significant design changes showed increases from 2 to 6 octane numbers over previous models.

5—The speed at which the majority of the cars of an individual make showed maximum requirement varied from 740 to 2000 rpm.

6—The average requirement of 32 cars that were tested under hill-climbing conditions was 2.4 octane numbers higher than the level road requirement.

This CRC report, number 280, contains 120 pages. It is available from the SAE Special Publications Department. Price: $4.00 to members, $8.00 to nonmembers.

Guideposts to AUTOMATION

If you have the urge to automate, to mechanize small lot production, or are trying to automate old machinery, here are some practical slants on things you'll want to consider.

Carl A. Augustine, chief draftsman, Bullard Co.

Based on secretary's report of Panel on "Automation and Its Effects on Manufacturing Methods," held as part of the SAE Aeronautic Production Forum, at the SAE Golden Anniversary Aeronautic Meeting, New York, April 21, 1955.

1. WHEN TO AUTOMATE . . . a matter of economics

Automation should be applied only when production requirements have reached the point where additional potential profits justify the expenditure for an automated line. The first step should always be a thorough comparative cost analysis of the present manufacturing method and the proposed method. Then manufacturing records should be reviewed to pinpoint where the highest cost is concentrated in the present system. This should be done by keeping a record of the amount of scrap; of the machining time of the various operations; of the tool costs and maintenance; of the handling between operations; and of accidents and other pertinent data. Consideration should also be given to the plant layout.

Much thought should be given to selecting the equipment for the automated line. "Will the design of the part to be automated remain stable, or will it change frequently?" should be asked and answered. In some cases, even though the design of the part remains stable for one and one-half years, the automation may pay off in approximately half this time.

To convince management of the need for automating a particular part, many points must be proved. Automation must save direct labor and cut labor fatigue. The expenditure must bring about an immediate dollar saving, greatly improve product quality, and cut spoilage.

You also need personnel skilled in electrical, hydraulic, and mechanical maintenance. A relatively small staff is required to maintain an automated line . . . only one electrician, one hydraulic maintenance man, and one mechanic for maintenance and, in most cases, only one operator to run the line. However, as the direct labor is reduced, the indirect labor is increased.

Floor space requirements are relatively small as compared with those of previous manufacturing methods. Scrap can be reduced to almost nothing by having a number of preset tools in ready reserve. However, replacements should be made at certain predetermined intervals; don't wait until tool breakage occurs.

Automation is not confined only to manufacturing procedures. For instance, foundries are automating for small castings. The old-fashioned method of limiting the amount of castings because of only being able to produce one heat a day has been replaced by a continuous molding, baking, and pouring process. An electric furnace can be charged in 20 min. Molding and core baking are done on a continuous belt and continuous pouring is done along the same line. Two furnaces can produce

10-15 tons, and small castings can be made at the rate of 20-30 per min.

Standarization in packaging raw materials and finished components by vendors or subcontractors to enable these components to fit better into the automated line is a prime requisite. This should also be urged upon raw materials vendors, so that castings, forgings, and stock sizes are uniform. Insistence upon this type of standarization will reduce maintenance costs.

In today's aircraft industry, certain ground rules and standards should be set up and applied with confidence. Design changes are constantly being made. There seems to be the risk that a given engine design may become obsolete before an automation line is half set up. For instance, in present-day jet-engine design, the large compressor casings are made of easily machinable aluminum castings. Designers are thinking of going to steel and ultimately to titanium. This will surely present difficult machining problems that can completely alter a mechanized line.

An automated line could be built in three or more sections, with enough conveyor space in between to allow for a bank of parts while one section is shut down for maintenance.

2. AUTOMATION FOR SMALL LOT PRODUCTION . . .

calls for new stance

Market stability as well as appropriate production requirements are necessary to sound automation for small lot production. Pertinent questions are: What is the market forecast? What is the potential business?

The Contributors . . .

. . . of the ideas in this article consisted of the following panel of specialists who are close to problems of automation.

Panel Leader

Walter A. Ehrlich,
chief draftsman, Bullard Co.

Panel Secretary

Carl A. Augustine,
chief draftsman, Bullard Co.

Panel Members

F. Steele Blackall III,
vice-president and assistant general manager, Taft-Pierce Mfg. Co.

Bernard Caine,
process engineer, Buick Motor Division, GMC

Leonard O. Carlsen,
chief engineer, Gleason Works

Everett Dennison,
manager of manufacturing engineering, Evendale Operating Department, General Electric Co.

Harold R. Foss,
director of manufacturing engineering office, Ford Motor Co.

Henry Maehl,
manager of production engineering, Fairchild Engine Division, Fairchild Engine and Airplane Co.

Lawrence Mahon,
industrial engineering manager, Wright Aeronautical Division, Curtiss-Wright Corp.

Complete report is available as SP-311 together with six other panels from Special Publications Department. Price: $1.50 to members and $3.50 to nonmembers.

For small lot production, the automated line should be designed so that as many as six or eight products can be manufactured with minimum changeover time.

For instance, although a month's supply may be run off on an automated line in a short time, it may be more economical to run off a six month's, or even a year's supply, before changing over to another product. But this presents two problems: (1) storage space for stock piling the present unused parts and (2) the possibility of engineering changes that would obsolete the stocked parts. A survey of the market trend would determine the risk in stock piling.

It would be an advantage to coordinate with the product designer the use of common parts in various type and size models.

A definite relationship should be fostered between process and design engineers. This would enhance the chances of making the design fit in with the automated line. Problems in manufacturing and automation facilities should be studied before making a major engineering change. The automated line should have flexible units so that design changes will not be hampered.

Small shops, to remain competitive, must mechanize to some extent. Here again, the extent to which they should mechanize depends mainly on the nature and volume of the product, whether it will conserve manpower and thus make a sizable cost reduction.

Handling and feeding devices should be used because they will make a manually operated machine almost completely automatic. Repeated manual operations, even though not physically difficult, will bring about operator fatigue and create an unconscious dissatisfaction with the job. Operators are more pleased with mechanized setups that require little or no exertion. An operator would rather be the "master" of two machines than to load, operate, check work, make adjustments, unload, and recheck or, in other words, be a "slave" to one that is manually operated.

Freeing the operator of loading and unloading eliminates accidents. Many annoying injuries, such as cut fingers and hands, are caused by sharp edges and burrs, through handling. The hazard of work dropping is eliminated by automatic loading, greatly reducing lost time due to injuries.

3. AUTOMATING EXISTING EQUIPMENT . . . doesn't always pay

Old equipment can be changed over to automation if close tolerances aren't required. This also depends upon what condition the old equipment is in. Will it pay to recondition or rebuild the old machines to bring them into shape to withstand the continuous flow of automation? Many old machines—10 or more years old—do not lend themselves to the continuous grind that automation demands. Most of the newer machines are designed with automation in mind. Therefore, it doesn't pay to automate old machines if they're worn out. However, machines not more than 10 years old, if in good condition, can be automated or mechanized by adding various type attachments.

Some of the newer equipment can be used to best advantage by installing it into a partially automated line. This can be done by installing work-handling devices, such as automatic loading and unloading attachments, to lessen operator fatigue. A system of preset tooling and preset gaging can be added to lessen down time caused by necessary tool changes. However, most old machines are not designed to cope with the new advances made in tooling. A large number of preventive maintenance personnel would be necessary because old equipment breaks down more often than new.

Single units can be mechanized by installing an automatic positioning device to eliminate jig and fixture handling.

Basic locating points should be developed into the product to insure accuracy, which otherwise would be lost where the product is transferred from one section to another. This will enable each part to be loaded into the machine in the same way. And tears, scratches, dents, and various defects from misalignment will be eliminated. Accidents and tool damage often are caused by improperly locating the part. Subsequent operations, such as finishing, painting, and plating can be simplified, eliminating such defects.

By means of electronic equipment, a toll can be made to remove a constant amount of stock. This is done with an electronic control that constantly reduces or increases the spindle speed. This has lengthened tool life considerably.

The National Machine Tool Builders Association, has completed an important study of chip breakers. This has measurably increased tool life and elimin-ted the problem of chip control.

The addition of tracer control, or contour controls, can turn a standard machine into a production machine, eliminating the repetitious measuring of the work.

Besides preset tool gaging, a system of automatic inspection units should be installed between the various operations. This automatically readjusts the parts that are not within the limits, so that the continuous flow is not interrupted. Also, these inspection units can be made to sort the parts into various classifications where the tolerances permit.

4. NEW DEVELOPMENTS . . . some here, more needed

Designers and builders of mechanized equipment, as well as the machine tool industry, have made giant strides in making their products easily adaptable to automation. They have successfully applied pendant and pushbutton controls, eliminating a maze of levers which can confuse even an experienced operator.

Many feel quicker ways of replacing worn out tools could be developed. It is true that the tool board has improved the problem. The board is equipped with timers that ring a bell or a buzzer, or sometimes flash a light, to warn the operator that it is time for a tool change. In connection with this board, gages are needed to preset the reserve tools so that they can readily be replaced.

This method has been a great improvement. However, a certain amount of shutdown time is required to make replacements. Perhaps an automatic method of immediately indexing another tool in place of the worn tool while the line is still running could be devised. Automatic gaging for gaging tool wear would have to be an integral part of this system. This could be called "built-in preventive maintenance."

Manufacturers of grinding equipment have introduced automatic means of redressing the grinding wheel at intervals to prevent glazing of the wheel. The machine automatically compensates for redressing or when a part dimension has gone beyond the prescribed tolerances.

Improvements could be made in loading and unloading devices because the part is often damaged when ejected from the machine. Parts are carelessly dropped into tote baskets and often the finish is marred. They become dented and even bent out of shape.

Experience has proved that automated lines should be mechanical as much as possible. It is true that mechanical applications can become cumbersome and difficult as compared to hydraulic applications. But the maintenance cost of hydraulic equipment far exceeds the cost of keeping mechanical parts in shape. Leakage, line breakage, scoring of pistons and cylinders all due to improperly filtered oil cause a large percentage of interruptions.

Temperature variations very often create a problem in maintaining size control. That's because of expansion caused by the heat generated by the continual running of the machine and hydraulic functioning. Cooling units can control the temperature of the lubricating oil and cutting compound, which would keep parts from going out of tolerance.

Many failures in an automated line are caused by electrical and electronic breakdowns. Use of low voltage controls would eliminate many failures, and replacement cost would be brought down.

What We Know About

Cam and Tappet

ENGINE builders know a lot more about cams and tappets now than they did a few years ago. But they haven't found any single, simple way to insure cam and tappet durability.

When cam and tappet failures became a subject for earnest study three or four years ago, it wasn't the first time the industry had met them. Cams and tappets are delicate, highly stressed parts that had given trouble before. But the materials and designs that cured the troubles in earlier engines aren't adequate for high-power overhead valve engines and the new oils compounded to keep them free of sludge and varnish.

The new engines have an increased demand for air as well as higher rpm. Top engine horsepower has been increased by lifting the valves higher to make way for more charge. Yet for satisfactory idle, valves must leave and land on the seat at exactly the right times. As a result, cam contours have to be steeper and sharper.

The steeper contours—together with higher engine speeds—result in higher valve spring opening loads. The combination of these factors increases the stress on cam and tappet faces markedly. (Paradoxically, it is the low speeds that cause trouble. At high speeds, inertia effects lower the contact stress at the critical point of dwell.)

In enough cases to worry engine builders, cam and tappet faces show failures that valve-gear experts classify into these three categories, shown in Fig. 1:

Wear—gradual smooth deterioration by friction.

Scuffing—deterioration by rapid surface roughening, welding, and tearing away of the surface metal.

Fatigue—chipping or break up under repeated cyclic stressing considerably below the ultimate strength of the material. Fatigue failure is also known as "spalling" or "flaking."

Failure may occur any time during the life of the engine. But most failures show up, several manufacturers report, between 5000 and 10,000 miles of operation. If cams and tappet faces survive 15,000 miles, chances are that they'll last the life of the engine.

When they do fail, the driver is likely to complain of valve gear noise, reduced engine power, engine misfire, backfiring through the intake system (when an intake valve is involved), or reduced top engine speed.

To end cam and tappet face failures, engine builders are seeking to find out (1) what combination of materials to use, (2) what shape to give contacting surfaces of cams and tappets, and (3) what types of lubricants do the best job on these highly loaded surfaces. Some of the facts that engine producers have uncovered so far on these three subjects are summarized below.

1. Materials

General Motors' Findings on Materials

The types of tappet failures observed in engines are primarily related to the tappet material.

Chilled cast-iron tappets ordinarily fail by fatigue, evidenced by spalling of the foot surface. The resultant roughened surface sometimes leads to damage of the cam with a consequent loss in engine performance.

Failure of the steel tappet can generally be classified as either wear or scuffing. Occasionally, a slight degree of spalling or pitting of the foot surface is also observed on this type of tappet. Scuffing, which appears as an irregular roughening of the foot sur-

1. Materials

2. Design

3. Lubrication

face, is by far the most serious type of failure, since it almost invariably results in severe wear of the mating cam.

Many carburized steel tappets, representing both engine tests and actual service in automobiles, were examined to determine the nature of the failures. Fig. 2 shows the foot surface of a typical scuffed tappet. Although the tappet wear was quite low (0.0008 in.), cam wear was excessive (0.0670 in.). Fig. 3 represents a cross-section of the foot surface of this tappet. It is evident that a pronounced change has occurred in the material adjacent to the surface, with the formation of a thin, light-etching surface layer and a dark-etching region immediately beneath it. The microstructure shown at the bottom of the figure is the normal structure ordinarily observed in this type of tappet.

The approximate hardness of the various regions, determined with a Bergsmann microhardness tester and converted to Rockwell C, is indicated on the figure. The appearance of the surface layer and its high hardness (Rockwell C 66) indicates that it is untempered martensite probably formed by rehardening of the original surface during operation in the engine. The minimum temperature at which such rehardening could occur, under equilibrium conditions, is approximately 1350 F. In the dark etching region, the temperature reached was not sufficient to produce rehardening, but the original martensitic structure was appreciably tempered.

As may be seen in the photomicrograph, spalling tends to originate in the rehardened layer. Some large white particles are also evident in the surface. These particles appear to be carbides, broken from the surface of the alloy cast-iron cam, which became embedded in the tappet foot surface during the momentary overheating. Fig. 4 shows the appearance of one of these rehardened areas on the actual foot surface of the tappet. The large em-

Fig. 1—Here's how wear, scuffing, and fatigue failures look in photographs and reduced photomicrographs.

THIS article is based on the following papers presented at the SAE Golden Anniversary Passenger Car, Body, and Materials Meeting, Detroit, March 2, 1955:

The Interrelationship of Design, Lubrication, and Metallurgy in Cam and Tappet Performance

by E. B. Etchells, Chevrolet Motor Division, GMC
R. F. Thomson, Research Laboratories Division, GMC
G. H. Robinson, Research Laboratories Division, GMC
G. K. Malone, Research Laboratories Division, GMC

Considerations Affecting the Life of Automotive Camshafts and Tappets

by M. F. Garwood, Chrysler Corp.
D. R. Kinker, Chrysler Corp.
J. J. Manganello, Chrysler Corp.

Camshaft-Tappet Metallurgy in Ford Overhead Valve Engines

by J. S. Laird, Manufacturing Staff, Ford Motor Co.
C. L. Stevens, Manufacturing Staff, Ford Motor Co.

Fitting the Tappet to the Camshaft

by V. L. Iles, Engineering Staff, Ford Motor Co.

All four papers will be published in full in the 1956 SAE Transactions. The General Motors and Chrysler papers are available individually from the SAE Special Publications Department at 35¢ each paper to SAE members and 60¢ to nonmembers. The two Ford papers are available in one cover at 35¢ for both papers to SAE members and 60¢ to nonmembers.

Additional information on cams and tappets (lifters) appeared in the article "Better Valve Lifter Life is Proving Tough to Attain" on pp. 40-44 of the December, 1954 SAE Journal.

Fig. 2—This steel tappet scuffed when run against an alloy cast-iron cam. Tappet wear was 0.0008 in. Cam wear was 0.0670 in.

bedded carbides are clearly visible. The foot surface was merely cleaned and etched to obtain this photomicrograph.

The rehardening observed on scuffed tappets is presumably the result of a momentary breakdown of the lubricating oil film, which permitted metal-to-metal contact with an attendant generation of heat. The severe cam wear observed in connection with scuffing would be expected with this type of failure, because the spalling of the rehardened layer produces a hard, rough surface which would tend to abrade the cam. The large carbides embedded in the rehardened material may also contribute to the cutting action.

Surface overheating of steel tappets does not always result in failure by scuffing and extreme cam wear. Fig. 5 shows a tappet which failed by wear, with little wear on the mating cam. Microexamination of the foot cross-section (Fig. 6) showed a shallow tempered layer at the surface. The surface temperature attained apparently was not enough to cause rehardening such as that observed on scuffed tappets, but the superficial tempering could presumably lower the wear resistance of this material enough to promote accelerated tappet wear. The reduced hardness produced by tempering would also result in a lowering of the fatigue endurance limit of the material and thus lead to spalling if the tempered material was not removed rapidly enough by wear.

The sensitivity of steel tappets to type of lubricating oil is not surprising in view of the susceptibility of this material to surface overheating. Slight variations in oil film conditions would be expected to produce profound differences in tappet operation.

Alloy cast iron tappets do not appear to be as prone to overheating as steel tappets. Scuffing of alloy iron tappets is observed only occasionally, usually under abnormally severe operating conditions, and alloy iron does not exhibit the sensitivity to type of lubricant which has been observed with steel tappets.

The compatibility of various camshaft and tappet materials has been investigated extensively. Since design and lubrication are such important factors, it has long been obvious there can be several cor-

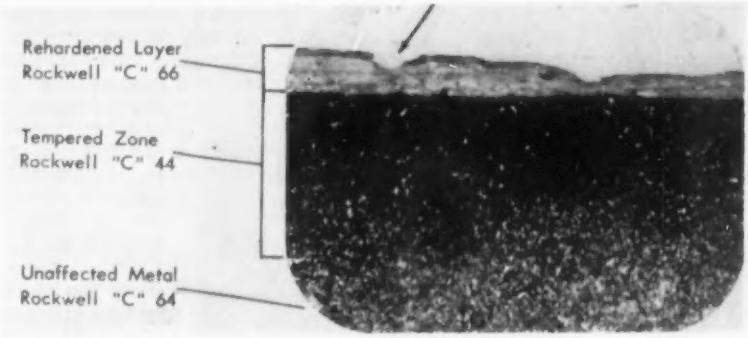


Fig. 3—The cross-section of the foot surface of the scuffed steel tappet shown in Fig. 2 exhibits a rehardened surface and the embedded material indicated by the arrow.

rect material combinations under different design and lubrication conditions.

Data from one engine point out some general trends of information:

1. Surface treatment improves the compatibility of materials.

2. Alloy cast-iron camshafts are generally more compatible with other materials than is SAE 1039 hardened steel.

3. An alloy iron camshaft will run with hardened steel tappets but a hardened steel shaft will not run with alloy iron tappets.

4. The incidence of spalling with chilled iron tappets on an alloy iron shaft increases with increased contact stress resulting from higher spring load and smaller radius of curvature.

Experience over a number of tests has shown a consistent variation in wear rate of cam lobes from position to position. This seems to be due to variations in microstructure from position to position. In general, the more cellular and less dense the carbide, the greater the incidence of failure. Unfortunately, this variation in microstructure is very difficult to eliminate in a production operation.

Ford's Findings on Materials

We are using carburized steel tappets with an alloy iron camshaft in the as-cast condition for our highest-production engines. This combination seems to be unique in the industry, but it does a good job for us.

It might be expected from the countless hours of experimenting and testing that have gone into past engine building, that one combination of materials would have been demonstrated to be the most desirable, and that the industry would have standardized on this combination. However, for camshafts, we still have carburized steel, alloy iron (usually hardened), and chilled iron; and for lifters, carburized steel, chilled iron, and hardened iron. A year ago, in their excellent SAE paper, "Wear, Scuffing, and Spalling in Passenger Car Engines," (appearing in the 1955 SAE Transactions) Ambrose and Taylor suggested the general adoption of hardenable iron

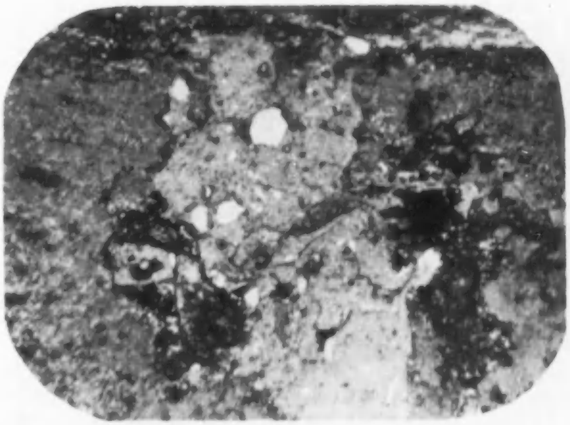


Fig. 4—In the rehardened area, the steel tappet foot shows spalling and embedded carbides (large white particles).

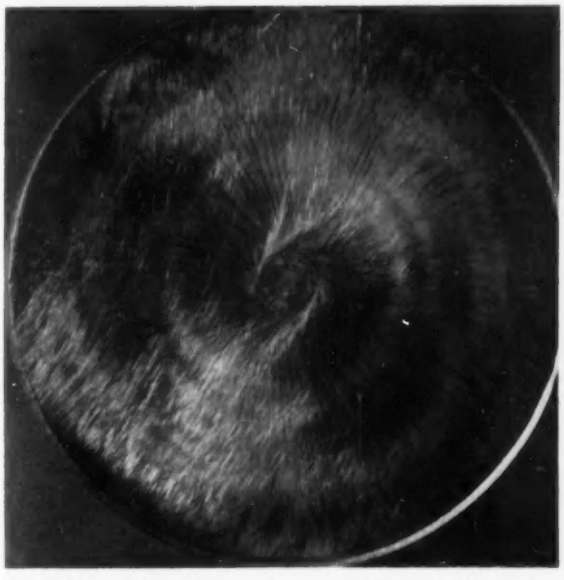


Fig. 5—This steel tappet wore when run against an alloy cast-iron cam. Tappet wear was 0.0073 in. Cam wear was 0.0008 in.

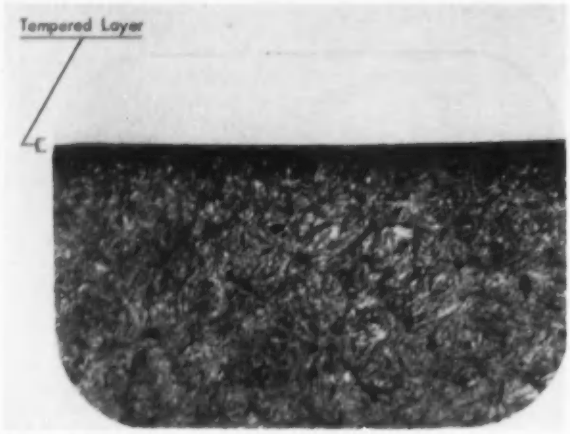


Fig. 6—The cross-section of the foot of the worn steel tappet shown in Fig. 5 exhibits a tempered surface.

for valve lifters, as being non-spalling, and permitting the addition of anti-wear agents to oils to minimize engine wear and scuffing. We cannot subscribe even to this degree of standardization. What is satisfactory for one engine may be completely unsuitable for another.

For a good many years, Ford has used cast-iron camshafts. Initially, a low-silicon iron, cast in green sand, was used. The sharp lobes of the camshaft tended to chill when cast in this iron. Extending into the sand, a thin tab on each lobe increased the rate of heat loss and insured that the lobe was chilled hard. Several types of tappets, varying from iron cast in permanent molds to carburized SAE 5120 steel, were used with these camshafts, with apparently equal success, the choice being based on economic factors rather than comparative performance.

The earlier camshafts were finished by grinding, so that hardness was no problem. Later changes in design made it necessary to do considerable machining on these cast camshafts. Since their sections and corners tended to chill to a hardness which made machining difficult, there were frequent complaints of flanges too hard to drill, and bearings too hard to finish. There were problems in the foundry also, as the analysis had to be controlled very closely and the chill tabs were a nuisance. If knocked off, the lobe tip was frequently shattered, whereas grinding them off was costly in labor and grinding wheels. It was suggested that the shafts be cast with soft lobes, which would later be hardened by flame or induction heating.

For this method, the low-silicon iron was not suitable, giving poor wear resistance when hardened. A hardenable alloy iron, similar to that used by Campbell, Wyant and Cannon, and possibly others, was adopted. The lobes of these camshafts were flame-hardened to Scleroscope 65, and, in L-head engines, had good wear characteristics with carburized and hardened SAE 5120 steel tappets. At first, these camshafts were cast in green sand. Later, they were produced by a shell-molding process. This process gave better dimensional accuracy with no noticeable change in the properties of the shafts after heat-treatment, or in their performance in engines.

The same type of camshaft was used during the development of the six-cylinder overhead-valve Ford engine, but this engine design called for production of an entirely different kind of tappet from that used in the L-head engines, namely, the mushroom type with a thin head and solid, slender body. Tappets of this type have commonly been made of chilled iron or heat-treated alloy iron castings. Our production facilities would be used to better advantage, however, if these tappets could be made from bar steel by upsetting. While a suitable upsetting machine was being developed, both hardened iron and carburized steel tappets were purchased for trial from outside sources. The hardened iron tappets were reasonably satisfactory, and showed distinctly better wear characteristics than the purchased steel tappets, but no better than those of the steel tappets ultimately produced in our own plant.

In our production, tappets are formed in a cold upset machine from ½-in. SAE 5120 steel rod. This fully automatic machine draws the rod to size, cuts to length, upsets and trims the head, coins a hemispherical recess in the foot, and rolls an undercut beneath the head. The heads are ground to a 60-in. radius with removal of not less than 0.020 in. of stock. In the upsetting process, folding or fracturing of fibers occurs to a depth of 0.010 to 0.015 in. Removal of 0.020 in. of stock insures the development of a solid face.

After grinding, the tappets are fed automatically and continuously into the retort of a rotary carburizing furnace. The retort rotates alternately to right and left, so that the stock moves forward for 2 min and 42 sec, and is then reversed for 2 min and 18 sec. In this way, the stock is constantly in motion while exposed to the carburizing gases, and passes through the retort in 7 hr. A temperature of 1700 F is held at the charge end, dropping to 1500 F at the discharge end, where the carburized stock is quenched in oil without exposure to the atmosphere. The carburizing gas consists of a mixture of endothermic carrier gas, propane, and ammonia, adjusted to give the desired microstructure in the carburized tappets.

The quenched tappets are drawn at 350–375 F. The stems are then ground to final dimensions. Heads are honed to a surface finish of 6 microin., only 0.002 to 0.003 in. of stock being so removed, so that the working face is held practically as carburized.

When carburized to an essentially eutectoid carbon concentration, these tappets gave only mediocre wear results with the hardened iron camshafts. Better results were obtained with higher carbon concentrations and microstructures showing islands, or nodules, of free cementite covering at least 50% of the surface.

While this tappet was being developed and subjected to merciless testing, the camshaft came in for further study. It was noted that camshafts on whose lobes there was a marked development of acicular carbide needles gave the best wear results.

Methods were worked out by which the development of this preferred structure could be controlled with considerable precision. Our success in this control appears to be due, in some degree, to the shell-molding process which had been adopted. The cool-

ing rate of the cams in shell molds appears to be more favorable to the development of the acicular carbides than that in green sand.

This development led to further study of the heat-treatment of these camshafts. How essential was this troublesome process? Could the quench be made less drastic, thereby decreasing the amount of warping which occurred?

Heat Treatment Studied

To obtain a base line for evaluation of the heat-treatment, some camshafts were run in the as-cast condition. To the surprise of all concerned, the non-heat-treated camshafts gave a good account of themselves. So long as the cams had the acicular cementite structure, the non-heat-treated cams ran at least as well as did the heat-treated cams with the carburized steel tappets. Results were not so favorable when the non-heat-treated shafts were run with hardened iron tappets, but various batches of the iron tappets showed little consistency, some showing very little wear, other batches showing considerable wear.

Extensive engine testing confirmed the desirability of the combination of an as-cast camshaft with carburized and hardened steel tappets. The as-cast camshaft also worked well with chilled cast-iron tappets in a larger overhead valve engine. Therefore, it was adopted for production, the heat-treating process was abandoned, and the equipment was made available for other uses.

The combination of an as-cast camshaft with hardened steel tappets containing a high percentage of cementite was used with success during a year's production of the six-cylinder overhead valve engine. During this period, development of the Ford eight-cylinder overhead valve engine was being concluded. The same tappets were used with as-cast camshafts in the prototype V8 engines. Performance of this combination was satisfactory in the extensive dynamometer and road testing of this engine and, accordingly, it was released for production.

After this engine was introduced, some camshaft distress was reported from the field at very early mileages. The extensive testing which had been done prior to production had not indicated the likelihood of such difficulties. Usually, only a single camshaft lobe was involved, but, in some cases, this lobe was wiped out completely in a few miles of operation, with the corresponding tappet pitted and spalled.

Failures were infrequent and could not be reproduced at will by normal engine operation, or even by a variety of abnormal conditions of operation. This indicated some uncontrolled variation in manufacture which produced a small percentage of abnormal parts.

The tappet faces were finely honed. The camshaft lobes were left rougher, and were etched by a phosphate treatment, since it had been found that lobes with a finish as fine as 10 microin., running against the 5 microin. tappets, had distinctly more tendency to gall than somewhat rougher cams. The phosphate treatment was found to be highly critical, and only a properly etched and coated cam would hold the film of lubricant so essential at break-in. Rigid controls were set up to insure that, on every camshaft, a coating 0.00012-in. thick was developed. The elimination of thin and spotty phosphate coating was highly beneficial, but did not, of itself, insure there would be no early failures.

No one cam appeared to be more susceptible to failure than any other. No relationship could be traced between the position of the cam with reference to gating, or to its position in the engine, and frequency of failure. However, it was noted that a high percentage of the tappets which had run against the galled cams had cases containing exceptionally high carbon concentrations.

As noted, in developing tappets to run against flame-hardened cams, a case containing a considerable amount of free cementite had been found desirable. This was continued with the as-cast shafts, the preferred structure being one in which 50 to 80% of the surface was covered with cementite. To produce this, the furnace had to be operated under extreme carburizing conditions, with 65 cu ft of propane per hr and 180 cu ft of ammonia per hr added to only 800 cu ft of endothermic carrier gas per hr. Control was very difficult, and the tappets produced showed considerable variations in structure from time to time. Carburization was most active immediately after the furnace had been cleaned and freed from deposited carbon. Under these conditions, the cementite nodules would coalesce into a solid white layer, 0.002–0.003 in. deep. It was noted that a high percentage of the early failures involved tappets showing this white layer.

Our knowledge of this white layer is incomplete. By metallographic tests, it is cementite, and this is confirmed by diffraction X-ray examination. If it consists of cementite (Fe_3C), the carbon content must be over 6%—a surprisingly high concentration. Since ammonia was introduced into the retort, the case may contain some nitrogen, although, at the carburizing temperatures used, iron nitride would be unstable, and little nitrogen would be expected to remain in the case. Some tappets with this structure have worn extremely well; a great many have been satisfactory, and some have been completely unsatisfactory. We have reached the conclusion that this structure is probably subject to variations in both composition and properties, and that, under the least favorable conditions, it may be very brittle. We failed to demonstrate this brittleness by impact tests, probably because the number tested was not large, and we did not happen to include a tappet of the very brittle type. It was obvious, however, that this structure was to be avoided as unreliable.

Test Procedure Developed

Up to this time, dynamometer testing had not produced wear patterns which could be considered comparable to those occurring in the field. This anomaly led to an intensive study of test conditions. Ultimately a test procedure was developed which gave results sufficiently consistent with field results to be considered significant. The test revealed a considerable variation in performance among the 500 tappets of September, 1953 vintage, used as a test lot.

A study was undertaken immediately to correlate the tappet performance variable with fits, clearances, material analysis, and heat-treatment.

Table 1—Cementite Classification of 33 Good-Performance Tappets

Class 1—25	Class 6—0
Class 2— 3	Class 7—1
Class 3— 2	Class 8—1
Class 4— 1	Class 9—0
Class 5— 0	Class 10—0

There seemed to be no single factor or simple combination of factors which would serve as a complete guide. However, it was noted that a fair correlation existed between tappet performance and cementite richness on the tappet face in about 60% of the cases.

The Metallurgical Department made reference photographs for cementite classification. Class 10 was predominately cementite (white), with small blotches of martensite (black). Class 0 was essentially martensite. September, 1953 tappets fell into Classes 6 to 10, this having been the desired production range resulting from tests on the 1952 6-cylinder engine. Contrary to 1952 data, however, the 1954 V8 engine gave better performance with Class 6 tappets than with Class 10 tappets, on a frequency basis.

Because frequency analysis was required to differentiate the general performance between Classes 6 and 10, it was acknowledged that there were variables in the production of 150,000 tappets per day which were not understood in terms of current mechanical and metallurgical knowledge. Hence, test data on small lots of tappets with non-production heat-treatment would be of questionable value.

It was decided to lower gradually the amount of cementite in production tappets, during which time Engineering Staff would test 100 tappets per day and Quality Control would inspect 15 production engines (after hot-test) per day.

At the stage in production where the reduction of cementite had resulted in a change from Class 10 to Class 1, 800 tappets were subjected to 11 hr and 40 min of cycle-testing. Following the tests, the tappets were visually screened, and 33 of the best-looking were checked for cementite classification. The results of this classification are shown in Table 1.

It is unfortunate that there was no explanation for the fact that most Class 7 and Class 8 tappets were poor while a few were good. However, the data indicated that production of Class 0 tappets was preferred over any higher-cementite range. Thirty days of continued observation of Class 0 tappets in production, and subsequent surveys of performance in the field have confirmed the choice of Class 0 tappets.

It was concluded that high carbon concentration in the case was not desirable for the Ford V8 engine; that a eutectoid case gave at least as good results as a higher carbon case; and that the easiest way of making sure that no tappets with the solid cementite layer would be produced, was to carburize only to a eutectoid composition. This was accomplished by drastically reducing the input of both propane and ammonia, with temperatures remaining the same. This results in a case which, after quenching, is essentially martensitic. Field results with the martensitic tappets and as-cast camshafts have been entirely satisfactory.

At this time, hardened iron tappets were tested in the V8 Engine with the as-cast camshaft. This combination was not satisfactory. There was a high rate of wear of both tappets and cam lobes. On the other hand, the steel tappets, carburized and hardened to a martensitic case, gave satisfactory wear with the as-cast camshaft in the six-cylinder engine as well as in the eight-cylinder engine. An adequate explanation of the apparently greater tolerance of the six-cylinder engine is still being developed.

The combination of as-cast alloy camshaft and martensitic tappet appears to be little sensitive to variations in oil. On the other hand, surface treatment of the camshaft must be carefully controlled. As noted, the camshafts are given a rigidly controlled phosphate treatment after complete machining. At assembly, the camshafts are carefully oiled, so that, when engines are started on the hot test, there is no possibility of even momentary dry contact between tappet and camshaft. Slight scuffing at this point can lead to ultimate failure.

Continued development has led to incorporation of copper in the iron analysis. This diminishes the tendency to slight pitting shown when as-cast camshafts with still steeper lobes are run against martensitic eutectoid steel tappets. The as-cast alloy camshaft is also being used with chilled iron tappets in a larger V8 engine with excellent results.

As cast, the camshafts present a good structure of acicular carbide in a matrix of fine pearlite, plus graphite. This structure, combining hard and soft materials, tends to adjust and heal over slight abrasions. If these camshafts are heat-treated, the pearlite is converted to hard martensite, which, once abraded, tends to cut and score to destruction. It is obvious that the use of camshafts as-cast is desirable from a purely economic standpoint. They work well in the engines described, but it is not to be expected that they will work well in all engines.

Chrysler's Findings on Materials

Materials changes offered the greatest benefits of any modifications of cams and tappets we considered.

We have done only a comparatively limited amount of testing on camshaft materials. Those tested include steel and higher-as-cast-hardness hardenable cast iron.

Steel camshafts were tested with chilled and hardenable cast iron and steel tappets. In combination with chilled cast-iron tappets, steel camshafts showed an increased resistance to scuff and wear and no tendency towards cam fatigue. Steel camshafts with hardenable cast iron tappets were unsatisfactory due to a greatly increased scuff tendency. However, the steel cams' high resistance to wear resulted in little lift loss.

Steel tappets on steel camshafts were also unsatisfactory due to increased scuff tendency. Again the amount of cam lobe wear was slight.

Preliminary investigations into the effects of increasing the as-cast hardness of the cam noses to Rockwell C 55 indicated this hardness would insure the presence of an abundance of primary carbides and consequently a camshaft that possessed better wear properties generally. This increase in hardness was obtained by adjustment of carbon-silicon ratio and increasing the chromium content. However, as expected, this increase in cam nose hardness caused a proportional increase in bearing and gear blank hardness to 302 to 341 Brinell. Machinability was intolerable without a change in the design of the camshaft.

Engine tests on the heat-treated experimental camshafts revealed that an increase in hardness of the cam nose increased the resistance to fatigue and wear but did not appreciably affect scuff tendencies.

In an effort to obtain indicated benefits, the foundry practice was adjusted to obtain camshafts on the higher side of hardness specification of 269 to 302 Brinell. This was accomplished by increasing chromium content from 0.80/0.90 to 1.00/1.10%.

Several unhardened camshafts were tested in engines. Although every effort was made to obtain the best possible structure in the cam nose, these tests showed such camshafts had greater fatiguing tendency than heat-treated cams tested under identical conditions.

No method was found to appreciably decrease scuff tendency. Further efforts to improve the camshaft metallurgically were concentrated, therefore, on heat-treatment of the cam nose. The production heat-treatment of the camshaft consists of flame heating the cam noses to approximately 1550/1600 F and oil quenching without subsequent tempering. If caution is not exercised, it is possible to overheat the nose. The overheating with which we were concerned was not only the actual melting of the metal, but also the heating of the cam nose in the temperature range of 1700 to 1800 F or above, which resulted in an undesirable microstructure. It had been proved that heating to this temperature, and subsequently oil quenching, produced appreciable amounts of retained austenite in the cam nose. The presence of retained austenite in the matrix lowers the nose hardness and makes it susceptible to failure by scuffing.

Hardenable cast iron was the only tappet material of approximately 15 materials tested which showed any improvement over standard chilled cast iron. The improved resistance to fatigue and scuff demonstrated in preliminary tests prompted a more extensive evaluation of the hardenable iron material, and a development program ultimately totaling 40 runs of an accelerated-failure test schedule and approximately 6000 hr of running followed. Approximately one out of 20 cycling test runs produced a failure within 40 hr, three out of 20 in 80 hr, and four out of 20 in 120 hr. Thus, only one out of five failed in 40 hr more running than that which failed four out of five chilled iron sets. Of the remaining four out of five hardenable iron runs made without failure, the majority ran over 120 hr with total test time running as high as 1000 hr on one run.

Tappets made of steel generally suffered scuff-type distress while iron materials other than hardenable iron were poorer due to increased fatigue tendency.

With hardenable iron tappets there appeared to be very excellent correlation between material structure and engine performance of tappet bodies.

Testing indicated that the as-cast hardness is directly proportional to susceptibility of the metal to fatigue. However, hardness was only important if it was due to the presence of carbides, and not the result of a very fine pearlite matrix. We have found that it is quite possible to meet the minimum hardness requirements of the specification (302 Brinell) and still have a material that is unsatisfactory because of carbide deficiency.

Permissible Hardness Bracketed

Engine tests conducted on high hardness tappet bodies (360 to 400 Brinell) also showed such tappets, whose microstructure approached that of chilled cast iron, had increased scuff failure tendencies. The tendency of low hardness tappets to fatigue and high hardness to scuff bracketed the permissible as-cast hardness range for satisfactory castings.

Samples of hardenable iron tappets in the hardness range of 302 to 341 Brinell were obtained and tested. Metallurgical examination indicated that these blanks possessed more carbides. Subsequently, the samples were subjected to engine tests with highly favorable results. Under the controlled conditions of this investigation, hardenable iron was definitely capable of withstanding more severe service than any of the other materials we examined.

Using test results as a basis, a material specification was established to provide a material for production that would have minimum tendencies to either fatigue or scuff. It was decided that the specifications should be as exacting as possible, but still realistic and practical.

Since engine tests indicated that primary carbides at the face were of utmost importance in the successful performance of a hardenable iron tappet, it was only logical that the material specification should be based essentially on microstructure. Most desired results dictated that the microstructure at the center of the face of the machined tappet body prior to hardening be a mixture of acicular-cellular carbides and graphite flakes in a matrix of pearlite.

The graphite flakes were specified as Type A or a mixture of types A and E. The allowable graphite size ranged from 5 to 7, as measured by ASTM specification A247 of latest issue. It was found that graphite flakes of Types B and D tended to permit early fatigue. Because of the importance of the amount and distribution of the carbides and graphite flakes on the performance of the tappet body, it was necessary to state that the end opposite the gated end of the cast blank be used as the face and have a hardness of 302 to 341 at the center. A maximum hardness of 321 Brinell was specified at the gated end. The face end hardness was indicated to insure satisfactory performance, while the gated end insured satisfactory machinability.

It was further decided that under no circumstances should the casting be heat-treated prior to machining. This restriction made certain the primary carbides would not be destroyed or altered in any way. Perhaps it is permissible to subcritically anneal the casting prior to machining without dam-

age to the primary carbides; however, it was believed the small gain in machinability would not compensate for the cost of such an operation.

Optimum heat-treatment for acceptable blanks was performed after rough machining according to the following schedule: Heat to 1575 F for 30 min, oil quench, temper for 1 hr at 300 to 325 F to a minimum hardness of Rockwell C 55. This minimum hardness requirement can be readily obtained, providing the cast blank possesses the specified carbide content.

Proper heat-treatment plays an important part in the performance of a tappet body. If the heat-treatment is not proper, failure is almost certain to occur. Improper heat-treatment includes decarburization, overheating, and underheating of the face. For example, investigations proved that if the atmosphere of the heat-treating furnace had a decarburizing potential, a breakdown of the carbide occurred. The carbides and graphite flakes on the face were destroyed, and this resulted in a tappet body that had scuffing tendencies similar to steel.

Difficulties can also be experienced as a result of improper flame and induction hardening. Such troubles are normally confined to overheating the tappet face, which produces an undesirable microstructure. Typical overheated structures have been determined to be partially dissolved carbide in a matrix of retained austenite and coarse martensite, a condition proved by engine tests to be very susceptible to scuffing.

2. Design

General Motors' Findings on Design

There are several practical methods of improving tappet-cam durability by design. Among them are:

1. Design for low contact stresses.

2. Avoid arrangements that allow mating parts to deflect and cause edge contact and high stresses difficult to calculate.

3. Design for low rubbing velocities with minimum overhang. This prevents edge contact.

4. Specify best finish and alignment possible consistent with production volume.

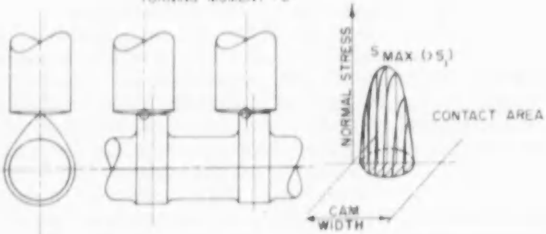


Fig. 7—Straight cam face with spherical-face tappets.

With flat-faced tappets, normal tappet rotation may be analyzed as the moment equal to the product of the offset and friction force. In the case of the cam and tappet face at a small angle due to deflection or misalignment, the moment is somewhat greater, contact area for oil film bulid-up is substantially less, and the tendency for oil throw-off is greater. All of this is conducive to generation of high stresses and consequent failure.

Spherical-face tappets will eliminate difficulties from deflection and misalignment providing the radius is not so large as to produce excessive spinning of the tappet and edge contact under adverse conditions, although the theoretical normal stress is higher than on the flat face tappet. The cams must have a slight taper, or the axis of tappet and cam must be deliberately tilted, to provide some rotation and partial oil film. (See Figs. 7 and 8.)

One important way of reducing contact stress (probably not well understood) is to provide for substantial radius of curvature on the cam at maximum stress point. There is a feeling in some quarters that a large nose radius is harmful due to tendency for increased scuff and wear. From a strictly stress computation standpoint, the minimum radius of curvature is not necessarily at the cam nose.

The distance from the axis of the tappet to point of contact with the cam is called the eccentricity. It is proportional to the velocity of the lift. It is obvious that the maximum eccentricity should be smaller than the tappet face radius. The use of (1) an offset to induce tappet rotation, (2) cams that are too wide, or (3) tappets with too large a spherical radius on the face when run with tapered cams, will often cause the resultant overhang to scrub the cam edge when excessive deflections or misalignments are present. Large eccentricities serve to distribute wear, but the rubbing velocity is increased.

Although rubbing velocity is ignored in computation of stress, its effect should be considered in the tests of a final design. Our original hydraulic tappet durability investigation indicated that this effect was appreciable. Of a total of 112 failures, it was found that 68.7% occurred at 2150 to 2550 rpm whereas only 31.3% were at 1350 rpm. (Approximately equal numbers of tests were run at each of the two levels of speed.) Rubbing velocity increases with increased engine rpm, while contact load tends to decrease at the higher speeds because of inertia effects. Since total stress is a function of rubbing velocity and cam load, it would appear that at some critical speed, the combination of each factor in some intermediate magnitude is more detrimental than the maximum of either.

With a maximum calculated contact stress on the order of 150,000 psi or more, it is imperative that surfaces be properly finished and as accurate as possible. Roughness and inaccuracies that rupture the borderline oil film will produce relatively higher stresses which may result in scuffing and rapid wear. Even though early scuff and wear do not show up due to improved materials, increased stress due to these factors which break the oil film, will result in spalling (or flaking) from shear fatigue. Often a simple operation of lapping both cam and tappet face edges will eliminate the tendency for scuff and rapid wear.

In overhead valve systems, other parts of the valve operating mechanism should be considered for finish, clearances, and compatibility of contacting surfaces, so that scuff and wear of these parts do not increase tappet face and cam contact loads with resultant higher stresses.

The value of phosphate type surface treatments to (a) inhibit scuffing at first contact, and (b) aid in the breaking-in process is well known. In addition, this process produces a pock-marked type of surface roughly similar in cross-section to hills and valleys, the hills having smooth plateaus. The oil carried in the valleys helps distribute the stresses of the partial or borderline oil film. The process must be rigidly controlled, and the initial surface to be treated must be smooth enough to carry the load. Otherwise early failure will result from scuff and wear.

However, representative tests on flat-faced chilled grey iron tappets and heat-treated forged steel flat-face cams have indicated that when calculated contact stresses exceed 150,000 psi, surface treatments alone will not insure satisfactory performance.

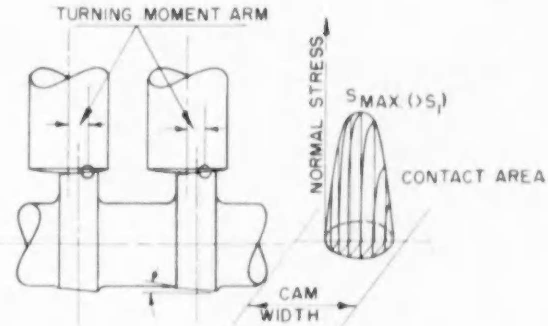


Fig. 8—Tapered cam face with spherical-face tappets.

Chrysler's Findings on Design

Tappet rotation impresses us as a great deterrent to cam and tappet deterioration. When tappets don't rotate, both the low-mileage and the high-mileage scuffing tendencies are bolstered. That is, cams and tappets are less likely to wear in satisfactorily during the first few hundred miles. And the cam is more likely to dig a "ditch" in the tappet if the combination survives to higher mileages.

Our standard tappet has a 30-in. spherical radius. We've tried modifying it to a flatter contour to reduce contact stresses, but the modifications haven't succeeded. Neither a 60-in.-spherical-radius cam nor a 30-in.-spherical-radius cam with ½-in. diameter flat on the center provided any benefit. A full-contact, flat-face cam cut scuffing and fatigue tendencies of the cam but induced fatigue failure in the tappet.

3. Lubrication

Ford's Findings on Lubrication

Because of current efforts to associate the tappet and cam wear problem with available types of lubricating oil, six oils were evaluated for comparative purposes. These various types resulted in measurable, but relatively small variations in tappet and cam lobe performance. A moderate change in cementite content of the tappets would offset the variation among the six oils tested.

Chrysler's Findings on Lubrication

We made a brief study of the effect produced by varying the amount of lubrication. In one type of test, the effect of preassembly lubrication was evaluated by oiling only one half of the camshaft by dipping it into the shell X-100 5W crankcase fill oil before installing in the engine. The other half of the camshaft was left dry. Approximately 10 engine test runs in which this procedure was followed, failed to show any distress pattern that would correlate with the oiling procedure. Failures occurred indiscriminately on the oiled and dry portions of the camshafts alike.

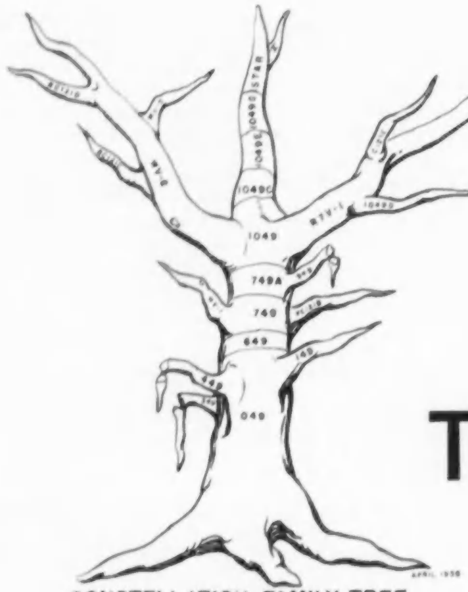
In a second type of test, the effect of increased lubrication during engine operation was evaluated. This was accomplished by means of an accessory oil manifold having individual "squirt" tubes aimed at each cam and tappet location so that the cam and tappet face areas were flooded with lubrication. Again no appreciable improvement in failure tendency was noted as a result of increased lubrication in the two test runs made.

In later more extensive studies, lubricant additives were found to be a very significant factor in cam-tappet distress, particularly when chilled cast iron tappets are employed. Certain additives were determined to be the cause of either high wear or corrosion fatigue. Hardenable cast iron tappets were affected very little by oil additives.

General Motors' Findings on Lubrication

Some time ago, when the automatic tappet was first investigated, it was noted that there was an increased tendency of the tappet face to spall. An increased tendency for the rocker arm bearings to score and push rod seats to wear was also observed. It was felt this was due to the spring in the tappet, which although light (10–12 lb) keeps the contacting parts together (essential for automatic operation) as opposed to the slack in the system 50% of the time with mechanical tappets, which allows for a more substantial oil film build-up.

Evaluation of a number of commercial oils by car tests has shown that crankcase oils vary considerably in ability to prevent wear and spalling of tappets. Differences in additive treatment appear to be more important in this respect than differences in viscosity. Exceptionally poor and exceptionally good oils could be distinguished, and the order of this rating was apparently not affected by changes in design or metallurgy. A more quantitative evaluation of the various oils, however, was not possible with car test data.



CONSTELLATION FAMILY TREE

The Constellation

THE first decade of Constellation service proves that it pays to introduce a bold advance in transport design, then let it sire an ever-improving line of descendants.

First orders for the Model 049 Constellation were booked right after VJ Day. Now, 10 years later, Lockheed is working on the Model 1649 Constellation designed to cross the Atlantic from east to west against winter winds without sacrifice of payload.

Even before VJ Day, the Constellation attracted attention by a record-smashing high-altitude pressurized flight from Los Angeles to Washington in 6 hr 57.5 min. That was on April 17, 1944. Compared to its nearest competitor, it had:

25% power advantage
24% speed advantage
24% range advantage
14% payload advantage
23% cost per seat mile per hour advantage.

In addition, it had:

The comfort of a pressurized cabin
The safety resulting from 52% reserve power
The safety of power boost on all control surfaces
The safety of high-lift Fowler flaps
The safety provided by a flight engineer.

Had the Constellation been a small increment in progress, it would have been killed off early by competition from new competitive models, whose designers were not timid, but merely late. We would never have recovered even our development costs, and our customers would have faced excessive losses through obsolescence — in other words, failure through timidity.

In contrast, Model 049 rewards came early. Immediately after VJ Day, in 1945, Lockheed booked orders from eight airlines for 103 Model 049's. These planes were soon delivered and set new standards in air travel and are, of course, still in use. Some of them have accumulated as much as 30,000 hr in the air with no limit yet discernible.

A hard, expensive, inescapable fact is that no matter how successful or advanced a transport design may be, it is necessary to start the design of its successor long before the sale of the first design has shown a profit. The designer or management who ignores this fact is not long for this industry.

As the war neared its end, the Wright Aeronautical Corp. had been working out the necessary engine improvements revealed by war service experience and was in a technical position to produce a new and truly commercial version of the 3350 engine. Interest in the Constellation grew with the evolution of the BD-1 engine. Accordingly, in May 1945 we began the design of the Model 649. Our objectives were:

1. To install improved equipment and accessories.
2. To provide more luxurious and quieter interiors.
3. To improve economy of operation by exploiting the increased engine power provided by the Wright BD-1 engine.
4. To improve the payload.

Eastern Air Lines was keenly interested in this airplane for medium-range operation. The airframe designed to receive the BD-1 engines and to provide some gross weight increase was accomplished, and 14 airplanes of this type were delivered to Eastern Air Lines.

Before the 649's had been delivered, it was new-model time in the design room. Both domestic and

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Excerpts from paper "The First Constellation Decade" presented at SAE Golden Anniversary Aeronautic Meeting, New York, April 19, 1955.

Family— and How It GREW!

foreign airlines began to show new interest in long-range operation. Our sales department cultivated this interest, while the engineering department developed outer wings with integral fuel tanks that would provide good transoceanic operation. This new model, designated the 749, was well received by domestic and foreign long-range operators. Fifty-nine were sold.

A major crisis always brings important facts to light and gives impetus to technical progress. The Constellation crisis was an electrical fire that culminated in a crash at Reading, Pa., in the summer of 1946, during a crew training flight. The Constellations were temporarily grounded, while detail changes in the electrical system were accomplished. The airplanes were then returned to service.

As a result of this experience, smoke evacuation procedures were developed for all transport aircraft. Had these procedures been known, this particular airplane could have been landed safely. On the Constellation, the established practice is to remove one exit over the wing and keep the cockpit sliding windows and crew door closed. Smoke from whatever cause is sucked out and the flight station remains clear.

In addition, fireproofing standards were established for cabin interior materials and insulation that are now complied with by all CAA-certificated commercial transport aircraft.

With the 749 back in service, we proceeded to review the stress analyses, landing gear reports, and drawings to work out the next step, that of increasing the allowable payload. By installing brakes of greater capacity, tires of increased ply rating,

SUPER CONSTELLATION, Model 1049G (left), as built today has wing-tip tanks which add 600 gal of fuel capacity at the expense of only 3-4 mph. Next model, 1649, will have new wing to accommodate all fuel without external tanks. Constellation as originally conceived in 1939 (right) had reverse-flow cowling, smooth windshield, round windows.

stronger main landing gear axles and shock strut cylinders, we were able to carry 5000 lb more payload than was then allowed on the 749. Thus the 749A came into being, with a take-off weight of 107,000 lb, landing weight of 89,500 lb, and zero fuel weight of 87,500 lb. Sales of the 749A continued; improvements such as the installation of jet stacks were accomplished to boost Constellation performance further.

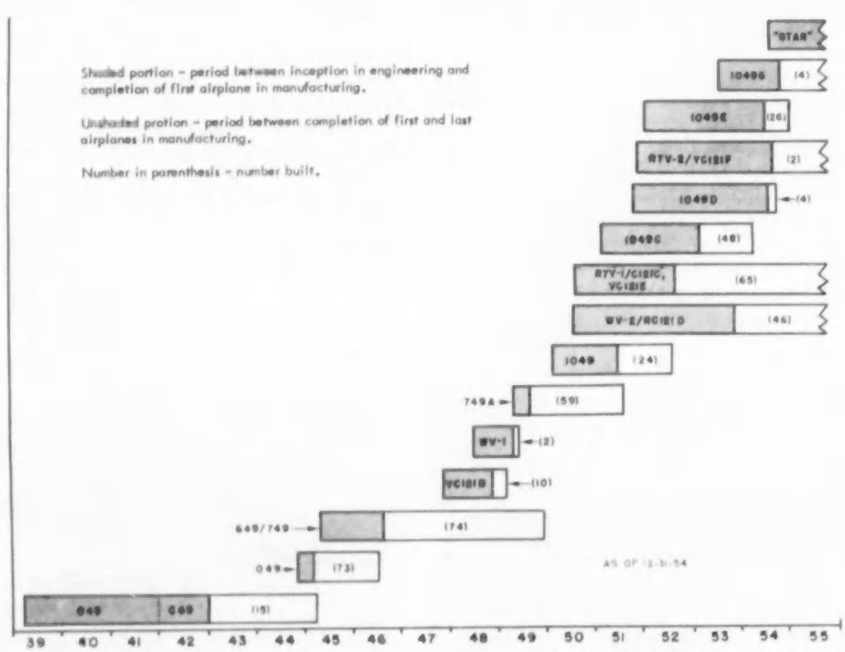
While sales of the Model 749A were continuing, we knew in late 1949 that the time was at hand to consider the design of a new version to meet new market conditions, which were becoming evident here and abroad. At that time many foreign operators were predicting that turbojet transports would soon be available. They were hesitant to stock up too heavily with the reciprocating engine variety.

To correctly assess the timing, the extent and the effect of the jet transport was a problem then and it remains a problem. I recall a Christmas card received that year, 1949, from a friend, a kindred soul of Paul Revere, who warned by showing *three* lanterns in the belfry of the Old North Church that the British were coming—with the Comet, that is—and soon. The question was, how soon!

While speculation and planning on the answer to this question were going on, there were certain immediate steps to be taken. Acceptance of air travel throughout the world had grown rapidly since VJ Day. More people would travel by air if they could afford to pay the price. The result was that the decision was taken to go forward with a new version—one that would carry more passengers at lower cost. Such a change required additional power and more room in the fuselage. The fuselage length was increased by 18 ft, 4 in., by adding barrel sections forward and aft of the wing in a manner that would maintain proper balance and stability. The Wright engine company developed an improved version of their 18-cylinder Cyclone, initially called the CA-1 and subsequently revised to the designation CB-1, which developed 2700 hp for take-off instead of 2500 hp previously obtained with the BD-1 on the Model 749A.

In addition to the increased passenger capacity, these airplanes included such improvements as rectangular windows, structural reinforcement for increased cruising speed, structural provisions for center section fuel tanks and the addition of more baggage loading doors. The first lot of the 1049 series was sold to Eastern Air Lines. TWA also ordered 1049's. The specific significant differences between the EAL and TWA airplanes were the interior features and, in the case of TWA, installation of center section fuel tanks with a capacity of 730 gal, instead of only structural provision for such tanks, as was the case for the EAL airplanes.

The Korean War and the A-bomb generated two vigorous branches to the Constellation family tree. The Navy needed an airplane that would carry either cargo or troops over long distances. Existing take-off, landing, and zero fuel weights were not adequate to accomplish the cargo and troop transport missions. The structural design of the Model 1049 wing had been stretched as far as practical. Further gross weight increases would require significant structural change to the wing. Further gross weight increases would also require more power. Being aware of the insatiable demand for



NEW MODEL is started before or very soon after completion of development on previous model.

more and still more power by the aircraft manufacturers, airlines, and military services, Wright Aeronautical had developed the Turbo Compound engine, which was the answer to our need for more power for these military versions of the Constellation.

Use of the compound engine brought with it a new problem—excessive exhaust flaming and afterburning out of the turbine exhaust hoods. The flames, in climb, before the engines were modified, carried back to the wing trailing edge. Structural strength was affected adversely until, after several months, hundreds of flight tests, and the expenditure of $2,000,000, solutions to this problem were found. No problem exists today with flaming, except that the passengers sometimes question the stewardess about the "fire in the engines."

This advance in powerplant design enabled us to go forward with the design of our next model—the R7V-1—to supply the requirement for a cargo-troop transport Connie. The structural design of the wing was changed to utilize 75 ST aluminum and integrally stiffened skin. This revision made it possible to raise the gross weight with a minimum increase of airplane weight empty. The fuselage was redesigned to provide large cargo doors in both front and aft sections of the fuselage. A floor development program was initiated that resulted in a configuration of extruded magnesium planks incorporating tie-down rings, seat attachments, and litter fittings. It was capable of carrying a unit loading of 300 psf, a value comparable to warehouse flooring practice. The whole floor assembly was sealed, so that it is possible to hose down the airplane interior. Object was, if the military were again faced with carrying out a Berlin Airlift type operation where such things as coal and flour were transported, it would be possible to keep the airplanes reasonably clean and avoid the accumulation of foreign matter in parts of the airplanes normally difficult to clean. The walls of the Navy cargo airplane are lined with glass fiber and equipped with rings for attaching cargo nets. This R7V-1 cargo Constellation was delivered to the Navy in quantity, with a take-off weight of 130,000 lb, landing weight of 110,000 lb, and zero fuel weight of 105,000 lb, with each Turbo Compound engine developing 3250 bhp at sea level for take-off. Subsequent lots of this model were delivered to the Navy at take-off weights of 133,000 lb.

The success of the R7V-1 with the Navy, and the appealing features of cargo-passenger convertibility of this model, had two results:

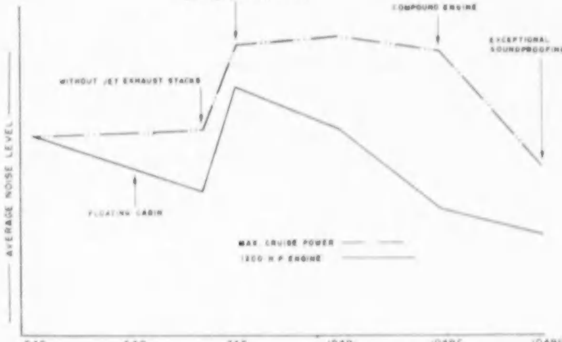
1. Air Force interest resulted in the sale of substantial numbers of a model similar to the R7V-1, but bearing the Air Force designation C-121C.

2. Commercial interest became apparent and resulted in the sale of convertible cargo-passenger transports to Seaboard & Western. These Constellations, designated the 1049D, have been in service largely on the trans-Atlantic run.

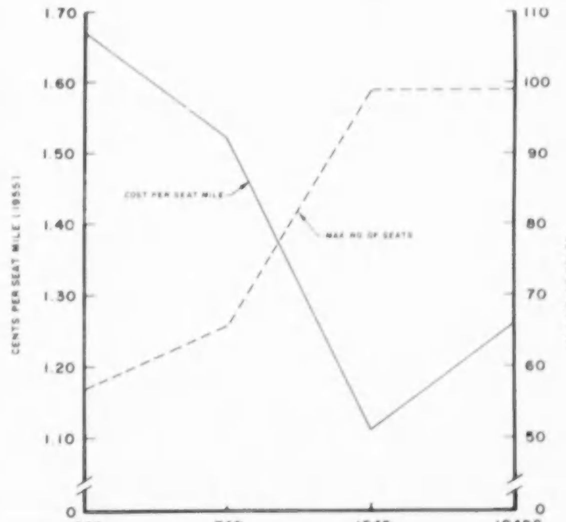
This brief sortie into domestic commercial cargo operation with the Connie stirs our enthusiasm and may portend further activity in the cargo field.

The second main branch of Constellation activity mentioned previously resulted from the placement of substantial orders to build radar early-warning airplanes. Development work had previously been carried on by Lockheed for the Bureau of Aeronautics on two modified 749A's, designated the WV-1. The new early-warning airplanes were called WV-2's.

The U. S. Air Force also placed orders for similar early-warning type aircraft, designated the RC-121. For both U. S. Military air services, the early-warning aircraft form the nucleus of eventual around-the-clock airborne vigilance of both coasts and ocean approaches to the North American continent. For such work, great range and endurance are essential. These airplanes carry approximately six tons of the latest radar and electronic equipment to high altitudes, thus surmounting limitations imposed by a curving horizon on straight-line transmission of unbending radar beams. Detection of



SUPER CONSTELLATION IS QUIETER than Model 049 despite Super Connie's much greater power



COST PER SEAT-MILE HAS FALLEN, in general. Higher cost for 1049G represents an increase in the level of luxury provided. Effects of inflation are excluded here

surface ships or aircraft is extended far beyond anything previously possible. The mission of the radar Super Constellation at the present time is primarily twofold—to provide early warning and to direct fighter or bomber interception.

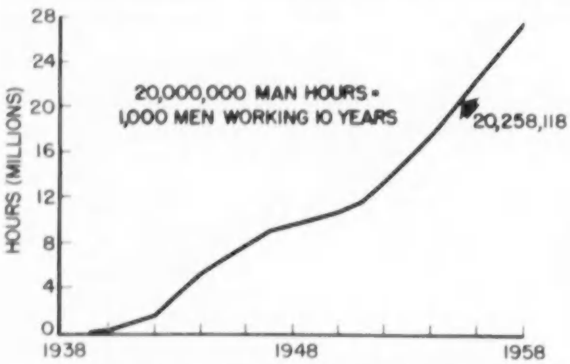
The fact that this craft serves as a stable platform for the early-warning function, in spite of the substantial protuberances affixed to the airframe, reaffirms our faith in the soundness of the original aerodynamic configuration. (Speaking of protuberances, these bulges have appeared so illogical to the layman that we have been suspected of designing an airplane to carry giraffes!) The lower radome, approximately 19 × 29 ft, on the underside of the fuselage center section, resembles a medium-sized swimming pool. The 8-ft-high dome on the top of the fuselage appears to be adequate for the upper extremity.

The triple tail configuration is particularly well adapted to accept these huge radomes, because the outboard verticals are not affected in any way by flow disturbances around the fuselage. These airplanes carry two 600-gal wing tip fuel tanks and a 1000-gal fuselage tank. Fuel capacity totals 8750 gal—2200 gal more than previous models. For early warning, these flying radar stations are the next best thing to a phone call from the enemy.

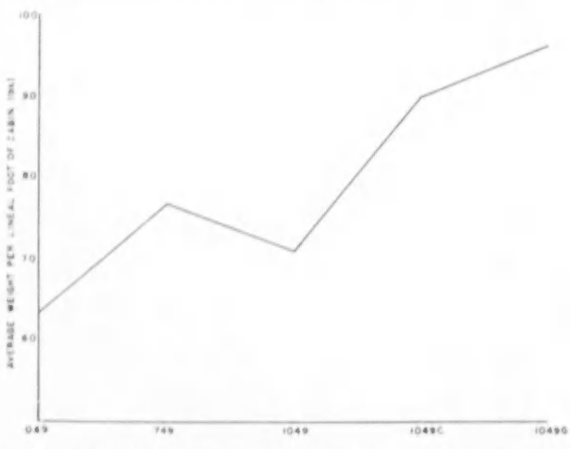
The increased power of the Wright Turbo Compound engine made the development of the commercial version of a Super Constellation to carry more load at improved speeds a logical next step. The new commercial configuration was called the 1049C, with a take-off weight of 133,000 lb, landing weight of 110,000 lb, and zero fuel weight of 103,500 lb.

The 1049C series was characterized by the design of luxury interiors. The services of the Henry Dreyfuss organization were obtained to help us evolve an interior superior in luxury appointments. Additional compartmentation was used that eliminated the "Holland-Midtown Tunnel effect" apparent in earlier airplanes. A new lounge concept was included which gave more flexibility in group arrangements, from romantic twosomes to gregarious canasta gatherings. Direct diffused lighting with controlled intensity and compartment switching was installed, in addition to the usual individual adjustable reading lights. Wood paneling was used extensively and interior details designed to achieve serviceability and pleasing appearance and a sense of roomy luxury in the compartments.

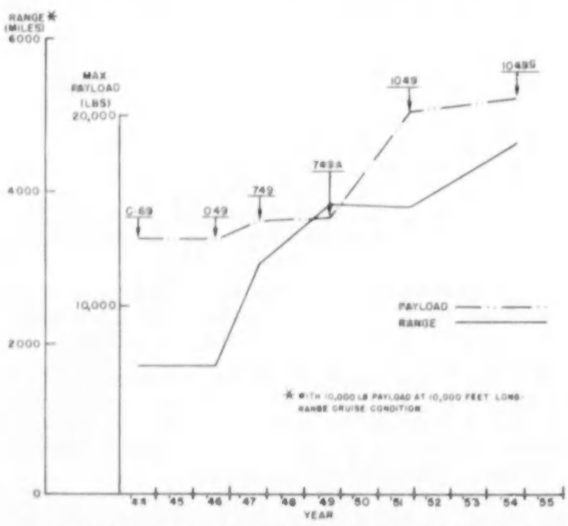
After the 1049C was engineered, we started through the same progressive cycle followed with the 749. The range was increased. Structural



HIGH LEVEL OF ENGINEERING ACTIVITY is required to design, redesign, and develop a basic series of large four-engined transports. At an average cost of $5 per man-hr, $100,000,000 will have been spent on the engineering aspects only for the Constellation series before 1956.



CABINS HAVE ADDED WEIGHT as they grew more elegant. The low value for the 1049 represents a standard interior minimum achieved just prior to the pronounced emphasis on luxury in the Dreyfuss-designed interiors of the 1049C and 1049G. Only about 25% of the increase is due to increased engine power and the added prop weight, structure stiffening, and sound proofing it entailed.



BOTH PAYLOAD AND RANGE HAVE INCREASED from model to model. Large improvement in range at inception of the Model 749 Constellation plane resulted from development of integral fuel tanks for the outer wing.

modifications were carried out to produce the Model 1049E, with a take-off weight of 135,400 lb; the landing and zero fuel weights remained the same as the 1049C, being 110,000 lb and 103,500 lb, respectively.

At this point, we were about as far as we could go without additional power. At this juncture Wright Aeronautical came forward with a revision to the DA-1 Turbo Compound engine, consisting of oil system changes, superchargers with improved impellers and shell cast diffusers for improved critical altitude performance, and an increase in meto power from 2600 to 2700 hp in low blower at 2600 rpm.

We have exploited this engine improvement in our Model 1049G, which has a take-off weight of 137,500 lb, a landing weight of 113,000 lb, and a zero fuel weight of 103,500 lb. And so the process goes on. We have already authorized the engineering of an increase in zero fuel weight to 108,000 lb, which will allow future "G" series Constellations to carry more payload and allow more flexibility in selection of the payload-range situation as it affects the airlines operating on different routes throughout the world. This increase in zero fuel weight is primarily useful to those operators whose route requirements dictate usage of a combination passenger and cargo interior—cargo in the forward cabin, passengers in the aft section. The interior design maintains luxury cabin appointments and provides effective cargo space.

This leap-frog process of payload-range increase and power improvement goes on. To provide still more range for Connie operators having special range requirements, we have adapted the tip tank installation developed for the early-warning radar airplanes to the commercial Connie. These 600-gal wing tip tanks are optional on our 1049 G series and are quickly removable and replaceable with standard tips, so that the varying route needs can be accommodated in an efficient manner. The airlines that have included tip tanks in their 1049G Constellations are TWA, NWA, KLM, and the German airline.

In 1952, with Navy support, we took advantage of the payload-range-speed possibilities of the Constellation equipped with the Pratt & Whitney T34 turboprop. This combination was designed for the Navy R7V-2, which is now flying, and is also used in the Air Force YC-121F.

Today's challenge and the goal of trans-Atlantic operators is east-west crossing against winter winds without sacrifice of payload. To meet this challenge, we are going forward on the Model 1649, which will have a wing with 12% more area, 33% increase in aspect ratio, and 46% greater fuel capacity. This model will utilize the latest development in the Turbo Compound engine, the EA-2, which develops 3400 hp for take-off and has a meto rating of 2800 hp. The propeller gear ratio is reduced to 0.35, so that tip speeds are reduced approximately 20% for quieter operation. By comparison with the 1049G, both the inboard and outboard engines are moved 5 ft 2 in. farther from the fuselage, to achieve reduced sound levels.

The new wing on the 1649 can well be the fundamental basic design for the weights, speeds, and fuel capacity required for turbine power.

The story of Constellation development is the story of striving continuously for improved safety, dependability, comfort, and economy; of overcoming limitations of time and distance and making available to an ever-increasing number of people the benefits of technological progress.

Errata

Two statements in the box on p. 22 of the July **SAE Journal** ("Rubber Springs Require Constant Level Device," by A. S. Krotz, J. H. Kramer, and R. E. Houser) should be amended as follows:

1. Rubber springs do not require independently sprung rear wheels. (Item 3 of the box erroneously indicates that they do.)

2. Changes in temperature will cause a vehicle suspended with rubber springs to vary as much as 15% in height. (Not 51%, as erroneously stated in item 1.)

How Combustion Chamber Design

Mechanical Octane= Anti-detonation Ability

THE efficiency of an engine can be increased by developing its mechanical octane number. This simply means designing the engine so that it can operate at higher compression ratios on fuels now available. In other words: increasing the borderline-knock power output per chemical (or fuel) octane number.

IGNITION control, valve timing, carburetion, engine-transmission relationship, and combustion chamber shape are the most important design factors that determine mechanical octane.

THESE are discussed fully in the original paper "Mechanical Octanes For Higher Efficiency" which will be published in 1956 SAE Transactions. This abridgment will limit itself to combustion chamber design.

PAPER is available in full from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.

ONE way of building more mechanical octane numbers into an engine is by designing the combustion chamber so that the rate of burning is speeded up. This completes the combustion process before end gas detonation can occur. A high rate of burning (giving a short combustion time) is controlled principally by flame travel, turbulence, and the mass distribution of the charge. These in turn depend upon the compactness of the clearance volume around the point of ignition.

Combustion Period Must Be Short

Actual combustion times of various chamber designs were calculated from indicator card pressure diagrams. Fig. 1 shows the results for two of the designs. Chamber A is a high octane requirement design. Chamber B is a low octane requirement design. The major differences are the charge for chamber B is ignited later than chamber A, and the rate of pressure rise is higher for design B.

Combustion chambers which have a low octane requirement exhibit somewhat higher end gas pressures and temperatures. This would be expected from the faster rates of burning. They show a substantially lower combustion time. Since higher pressures and temperatures tend to promote knock rather than suppress it, combustion time must be the dominant controlling factor which allows a reduction in octane requirements to be obtained through combustion chamber design. An associated controlling factor is flame travel.

Flame Travel Influences Combustion Rate

Flame travel length is measured from the point of ignition to the most remote point in the chamber with the piston in the position it actually occupies when the last part of the charge is burned. Moving the spark plug toward the center of the bore, which effectively shortens flame travel and also shortens the combustion period, results in a reduction in fuel requirement. This design change also increases the borderline knock power output on a gasoline which knocks below the peak power spark setting.

The per cent of the mass of the charge which is burned at any instant during combustion was calculated for several representative experimental chambers. The results show that, depending upon the total combustion time, 90–99% of the charge is burned with the piston moving on the downstroke at the beginning of the expansion process. This is illustrated by Fig. 2 which shows the per cent of

Affects Mechanical Octanes

Compact,

Highly Turbulent,

Fast-Burn Combustion Chambers . . .

. . . will increase Mechanical Octane.

D. F. Caris, B. J. Mitchell, A. D. McDuffie, and F. A. Wyczalek,
Research Laboratories Division, GMC

Excerpts from paper "Mechanical Octanes For Higher Efficiency" presented at the SAE Golden Anniversary Summer Meeting, Atlantic City, June 13, 1955. This paper will be published in full in 1956 SAE Transactions.

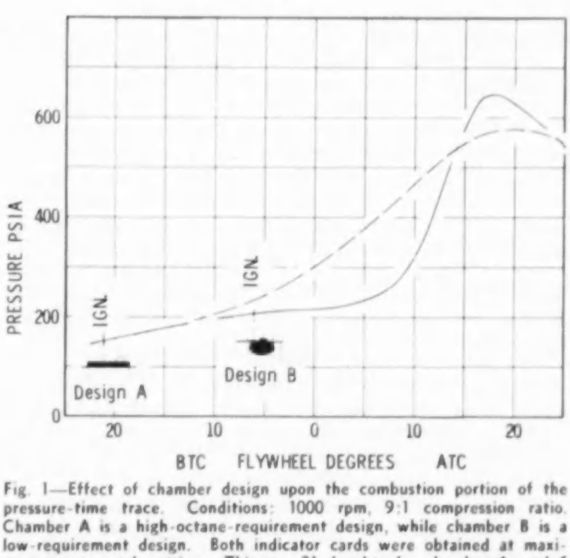


Fig. 1—Effect of chamber design upon the combustion portion of the pressure-time trace. Conditions: 1000 rpm, 9:1 compression ratio. Chamber A is a high-octane-requirement design, while chamber B is a low-requirement design. Both indicator cards were obtained at maximum-power spark setting. This was 21 deg btc for chamber A, and 6 deg btc for chamber B. Major differences: In chamber B, charge ignites later and rate of pressure is higher.

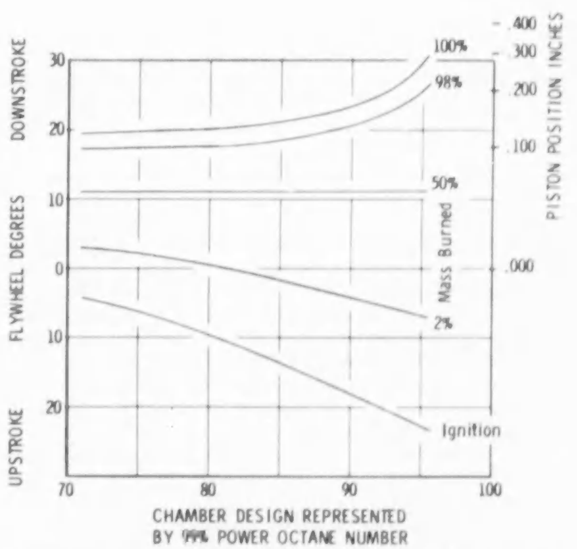


Fig. 2—Between 90 and 99% of the charge is burned with the piston moving on the downstroke at the beginning of the expansion process. This graph shows the effect of combustion chamber design (as represented by octane number) upon combustion period and piston position at any instant during combustion.

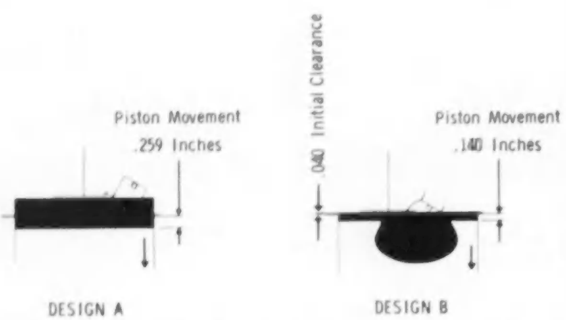


Fig. 3—Piston position at the instant when the last part of the charge has burned. The flame front must burn out to the cylinder wall if all of the charge is to be consumed.

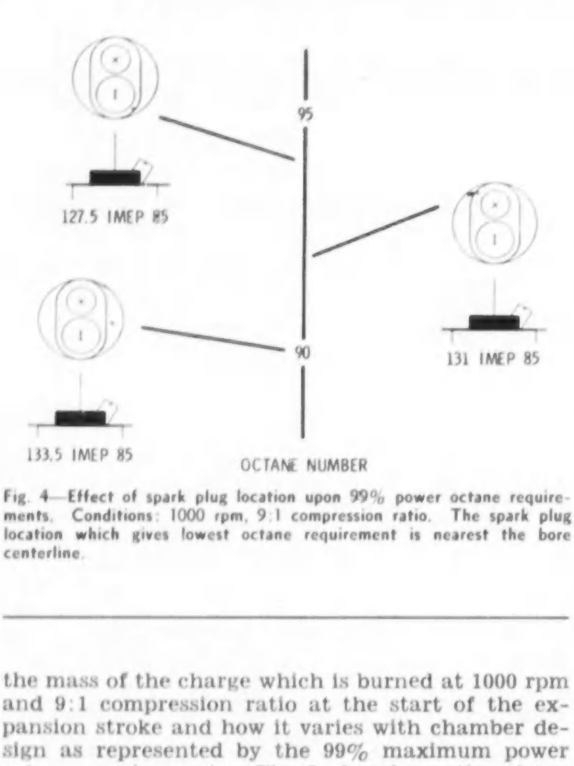


Fig. 4—Effect of spark plug location upon 99% power octane requirements. Conditions: 1000 rpm, 9:1 compression ratio. The spark plug location which gives lowest octane requirement is nearest the bore centerline.

the mass of the charge which is burned at 1000 rpm and 9:1 compression ratio at the start of the expansion stroke and how it varies with chamber design as represented by the 99% maximum power octane requirements. Fig. 2 also shows the piston position at any instant during combustion.

By the time the last 2% of the charge is being consumed, even in the case of chambers with high rates of burn, the cylinder wall is exposed so that it becomes the actual confining limit of the combustion space. This is illustrated clearly for two chambers by Fig. 3. Design A is a slow burning high octane requirement design, while design B is a fast burning low octane chamber. Both chambers are shown with the pistons on the downward stroke at the end of the combustion process. The dimensions in inches show the calculated piston position at the instant when the last part of the charge has burned, which is also the time when the flame front has just reached the left hand cylinder wall.

It is quite apparent from this illustration based upon combustion periods calculated from balanced diaphragm pressure cards that the cylinder wall will always be the confining limit of the combustion chamber. The flame front must burn out to this boundary if all of the charge is to be consumed.

Fig. 4 shows the effect of spark plug location for a design which has moderate piston coverage. The plug location which results in the lowest octane requirement is nearest the cylinder bore centerline.

To be certain that the improvement in fuel requirement has not been obtained at a sacrifice of power, the borderline knock imep using a regular gasoline of 85/75 octane is shown beneath each design. The borderline knock power is highest for the chamber which has the lowest fuel requirement.

Fig. 5 shows the results of a study of the effect of spark plug location conducted during the development of the 1954 Buick combustion chamber. The central plug location near the bore center results in a substantial reduction in octane requirements. The last chamber with the three plugs firing simultaneously results in an additional improvement. Once again, examination of the power output on a regular grade fuel shows the highest value for the chamber with the lowest fuel requirement.

Turbulence Increases Combustion Rate

Turbulence is another important influence on the rate of combustion and mechanical octane number. It is present in varying degrees during the intake, the compression, and the combustion periods. Best results are obtained by arranging the clearance volume at the point of ignition so that the largest possible percentage of the mass of the charge will be concentrated at this point. Also the clearance between the piston and the cylinder head should be held to a minimum so as to use piston coverage effectively.

Turbulence can be achieved by properly directing the intake passage or shrouding the intake valve on one side so that mixture flow is blocked in one direction and forced to swirl in the other direction. From a practical standpoint it would be difficult to use shrouding to influence the fuel requirements at low engine speeds where knock is a problem and still allow normal breathing at high engine speeds where power is desired. For example, the effect upon breathing of shrouding the intake valve is shown by Fig. 6. This chart shows combustion pressures for three cases: a normal intake valve which allows mixture flow in all radial directions, a valve with a 90-deg shroud, and a valve with a 180-deg shroud.

By comparing compression pressures of both shrouded valves to the normal valve, it is apparent that this method of reducing the fuel requirements restricts breathing above 1800 rpm. At speeds below 1800 rpm, breathing is normal as indicated by identical compression pressure values.

However, in order to explore this approach and also as a means of determining the potential effects of the intake turbulence factor, without regard for practical applications of the system, a shrouded intake valve was installed in a combustion chamber

and tests were run only at 1000 rpm where breathing is still normal.

Fig. 7 shows the effects of the intake shrouds described above for a combustion chamber with no piston coverage. The shrouds were fixed to direct the incoming charge toward the exhaust valve.

A substantial reduction in octane requirement results as the degree of shrouding is increased. A comparison of borderline knock power on 90 octane fuel shows a higher value for the valve with the 180-deg shroud.

The direction of the intake charge is very influential in determining the extent of the reduction in octane requirements. The intake valve was fixed in the desired positions so that the shrouds would be located as shown by Fig. 8. In every case, shrouding results in an improvement over the unshrouded intake system regardless of direction. A comparison of borderline knock power on 70 octane fuel shows that power has not been sacrificed to obtain the gain in fuel requirement.

In summary, turbulence produced by shrouding intake valves is very effective in reducing the fuel requirements. Therefore, the effects of turbu-

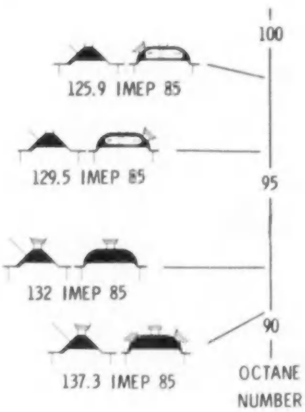


Fig. 5—Spark plug located near the bore center results in lower octane requirements. Three plugs firing simultaneously give even better performance.

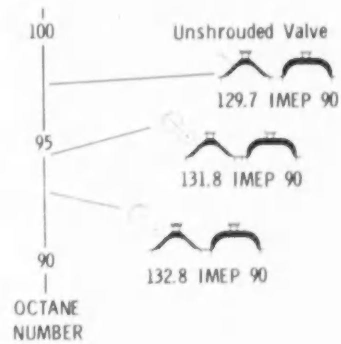


Fig. 7—At 1000 rpm, shrouding will reduce octane requirement.

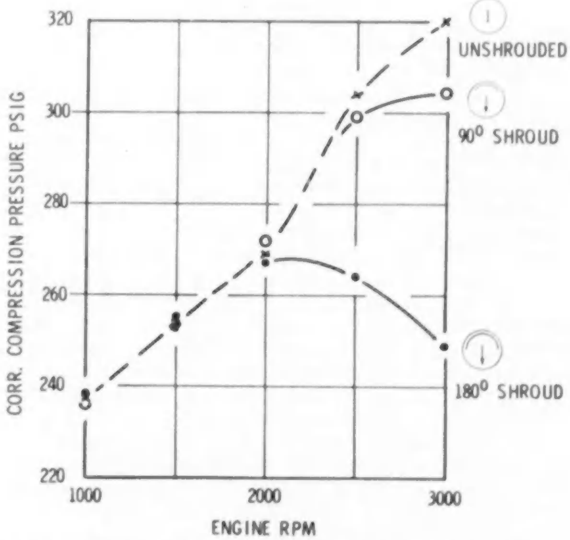


Fig. 6—Shrouding intake valves restricts breathing above 1800 rpm.

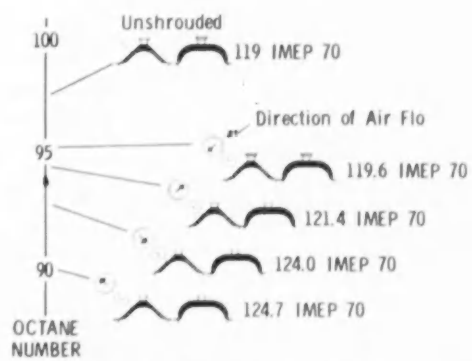


Fig. 8—Effect of direction of intake valve shrouding upon 99% power octane requirements at 1000 rpm, 9:1 compression ratio.

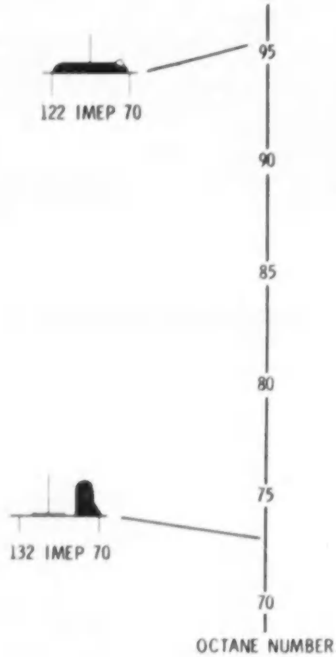


Fig. 9—Effect of piston coverage and mass distribution of the charge upon 99% power octane requirements at 1000 rpm, 9:1 compression ratio.

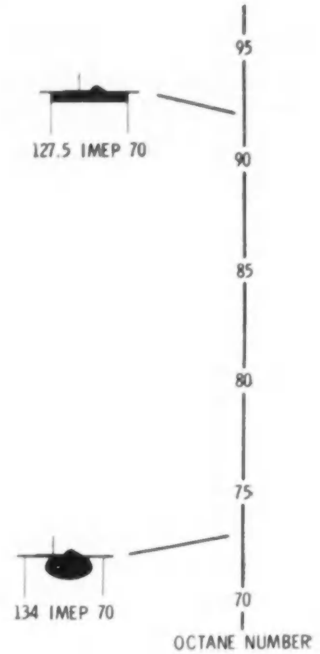


Fig. 10—Piston design distributes clearance volume around the spark plug, concentrating as much of the charge as possible at the point of ignition. The better the concentration, the lower the octane requirements.

lence which are produced at this early stage in the engine cycle must persist throughout the combustion period to produce the results observed. However, this method also restricted engine operation above 1800 rpm when the shrouding was increased to the point necessary to produce a measurable effect upon fuel requirements. Directed intake ports are not likely to be effective in producing turbulence at low engine speed where knock is a problem, unless the port areas are small enough to produce the necessary high velocities. Small port areas are impractical because of high-speed breathing requirements.

Although the designer is somewhat restricted in the gains he can achieve by means of intake turbulence, fortunately a more practical approach is available. Piston coverage has the effect of physically displacing the charge from one side of the chamber to another as the piston approaches top dead center. To accomplish this, the piston is shaped so that part of it matches the head at top dead center with only enough clearance to prevent interference.

It is generally believed that this physical arrangement of the chamber results in compressive turbulence, or "squish."

Presumably, the amount of compressive turbulence is proportional to the per cent coverage. However, no attempt was made to establish a relationship between turbulence and coverage by measuring mixture velocities. It was found that although the fuel requirements are reduced as per cent coverage is increased, the mass distribution of the charge is also an essential parameter.

Mass Distribution

Proper mass distribution is obtained by concentrating most of the clearance volume around the spark plug. Combined with piston coverage, this influences fuel requirement effectively, since its beneficial effects occur just before and during ignition of the charge. Also, it doesn't limit normal breathing or functioning of the engine as intake shrouds or angled intake ports do. Fig. 9 shows the effect of piston coverage with proper mass distribution. Note that both chamber designs have the same distance from point of ignition to the cylinder bore. Therefore, the reduction of 23 numbers in octane requirement must be a result of piston coverage and the manner in which the greater part of the charge is concentrated around the point of ignition. It is quite apparent that physical change in chamber design actually is the most important single design change which can be used to produce an improvement in fuel requirements. Examination of the borderline-knock-limited power values obtained using 70 octane fuel shows that the lower octane requirement chamber produces 8% more knock-limited power than the high octane requirement design.

Another example of the beneficial effects of high piston coverage is shown by Fig. 10. In this example the spark plug location is identical for both

chambers because the same cylinder head was used. Therefore, flame travel distances to the cylinder bore are the same in both cases. The design of the piston, which distributes the clearance volume around the spark plug concentrating as much of the mass of the charge as possible at the point of ignition, together with the high piston coverage which results from this arrangement, illustrates the ultimate objective of chamber design. This piston design change produces a 19 octane reduction in fuel requirements. A check of the knock-limited power on 70 octane fuel shows a 5% improvement in output.

Another design variable which produces a profound effect upon octane requirement is the clearance between the head and piston at top dead center. This is often referred to as "quench thickness," implying that quenching is the beneficial mechanism involved. However, it is very difficult to separate the "quench" effect from the other factors involved. To evaluate this factor properly, quenching must be varied while all other variables are held constant. Fig. 11 shows the two designs which are identical except for the clearance between the head and piston. Reducing the clearance results in a 12 number improvement in octane requirement with a corresponding increase in borderline knock power on 70 octane fuel.

It is difficult to understand how such a minor change in design can have such a large effect. Since the piston is on the down stroke when the last part of the charge is burning, quenching action at this time is not very likely to be effective. Therefore, a reduction in clearance must result in a substantial increase in turbulence and in increased borderline knock output. The improvement is attributed to increased "squish" velocities resulting as the piston approaches top dead center.

Another factor is the rapid change in mass distribution of the last part of the charge to burn due to piston motion. As the piston moves down, the flame front is accelerated toward the cylinder wall, resulting in shorter combustion time. Both increased piston coverage and reduced quench thickness accentuate this effect.

Fig. 12 shows the effects of other combustion chamber designs. The small design differences give remarkable reduction in octane requirement. In both designs the cylinder heads and the piston clearance at top dead center are identical. Therefore, the difference in construction of the piston must result in a substantial increase in the utilization of turbulence and better mass distribution of the charge. Examination of the borderline knock-limited power values shows an increase in power on 70 octane fuel for the low octane requirement case.

Deposits Increase Octane Requirement

Deposits on combustion chamber surfaces increase the compression ratio, decrease power output, and increase fuel octane requirement. The compression ratio increase is a result of the decrease in clearance volume due to deposits. The decrease in power, in spite of the increase in compression ratio, is due to a reduction in charge density. But, although deposits increase the octane requirement of each chamber design, the chamber shape which has the lowest requirement when clean still has the lowest requirement with deposits.

Combustion chamber design also determines, to some extent, the type of deposits which are formed, which in turn, affect surface ignition.

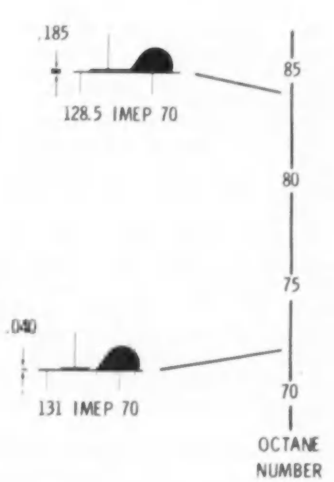


Fig. 11—Reducing the clearance between the cylinder head and the piston will reduce octane requirement 12 numbers.

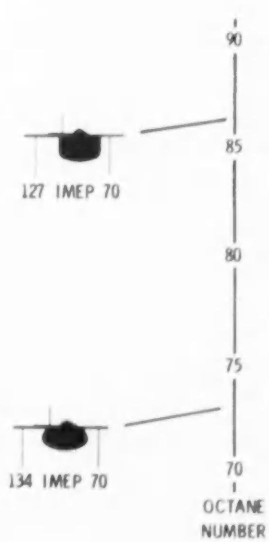


Fig. 12—Difference in piston design influences turbulence and mass distribution. Clearance between cylinder head and piston is identical with these two designs.

How to Package Against

THIS article is based on papers presented at the symposium "Packaging Against Shock and Vibration" which was held during the SAE Golden Anniversary Aeronautic Meeting, New York City, Monday, April 18, 1955. The symposium chairman was W. L. Hardy, Foster D. Snell Co. H. J. Wenborn, General Electric Co., and E. H. Jones, Bell Telephone Laboratories, were chairmen of the morning and afternoon sessions, respectively. S. M. Birnbaum, Wright Air Development Center, was secretary.

The following papers were presented at the symposium:

Testing For Instrument Fragility
J. T. Muller,
Sperry Gyroscope Co.

Fragility Rating
W. H. Skidmore,
Weston Electrical Instrument Corp.

Cushion Design from Fragility Rating
H. E. Nietsch,
Robinson Aviation Inc.

Air Cargo Shock Damage Experience in Handling and Shipping
A. C. Botsford,
American Airlines, Inc.

Development of Shock and Vibration Test Methods for Packaging
John Cammarata and Vincent Atalese,
Arma Corp.

Flexible Suspension Systems for Equipment in Transit
J. J. Goodill,
Lord Mfg. Co.

Practical Development of Suspension Cushing for a Guided Missile
R. L. Wiltse,
Chrysler Corp.

The papers are available in multilith form from SAE Special Publications Department, price: 35¢ (each paper) to members, 60¢ to nonmembers.

THE amount of protection which a piece of equipment needs while being shipped or stored depends upon its fragility. If the container is not strong enough or does not have enough cushion material, it will not protect the equipment from damage caused by dropping or crushing. If the packaging is the wrong type, damage is often caused by the equipment vibrating in resonance with external shipping vibrations, or with the vibrating container itself. If the equipment is over-protected, it will arrive safely but at needless cost and wasted shipping space.

Until recently nothing much was known about efficient packaging. But the need for protecting electronic equipment for aircraft and guided missiles has given great impetus to investigating packaging problems. (See SAE JOURNAL, March, 1955, pp 53–57)

An attempt is being made to establish a "fragility rating" — that is, a standard classification system that specifies the largest amount of likely accidents and their nature that a piece of equipment can tolerate without damage. Once this rating is available, proper packaging can be provided that is both adequately protective and economical.

One fragility index that has been suggested consists of two parts: (1) the maximum deceleration which the item can tolerate without damage, and (2) the natural frequencies (below 75 cps) of any component of the item.

The deceleration portion, expressed in *gs*, governs the very minimum cushion design. The resonance portion is, in effect, a statement that the package should be designed so as *not* to vibrate at any of these frequencies for fear of damaging the resonating component by amplification of resonance.

These two parts of the rating are determined by shock and vibration tests of the equipment in a laboratory.

Once the fragility rating has been determined, the type of cushion material, shock mounting system, and package case can be selected or designed.

It is not necessary to wait until the equipment has been manufactured and successfully tested to design a package. As soon as the physical dimensions

Shock and Vibration

of the item have been set, a package should be designed so that the completed container and packing may be tested at the same time as the equipment.

The equipment is scanned for frequency vibration to a point estimated by good engineering judgment to be above the critical range. Then a low pass filter up to this point is used during the shock test.

In determining the character of the shock pulse which is transmitted to the equipment through the container, it is not necessary to jeopardize the actual equipment. A mock-up can be constructed duplicating the center of gravity and the major dimensions of the equipment.

Then three accelerometers, each parallel to one major axis, are rigidly mounted, and a recording system set up to measure the filtered accelerometer signals.

The test is made and repeated for each corner and side as called for in the packaging specifications. It is easy to determine, from the simplified shock waves, the most pertinent frequencies and the magnitudes of the accelerations at each point.

The shock record will show whether or not actual or virtual bottoming has occurred (by a marked discontinuity or additional pulse at some time interval after the initial velocity change).

By comparing the spectrum obtained during the vibration frequency scan with these major shock pulse frequencies, the probability of damage to the unit due to resonance can be estimated.

If bottoming does not occur and troublesome frequency components do not exist, then the attenuation achieved on the package is apparent from the magnitudes recorded.

If bottoming has occured, it is obvious that to avoid damage to equipment, the package must be redesigned to afford greater travel, or the cushion material must be changed to reduce travel after shock.

If high energy components of the shock include frequencies very close to the natural frequencies of the equipment then the package and cushion should be modified. Certainly advantage should be taken of the ability of the equipment to withstand some shock. If it is found that a reasonable safety margin exists between the magnitude of the pulse transmitted to the package (with no frequencies disturbing the equipment) and the shock for which the equipment is designed, the package cushioning may be reduced to get a lighter, smaller, and less expensive package.

The vibration characteristics of the package can be calculated from the natural frequencies of the cushion materials and a piece of equipment of known weight based on the load-deflection curve of the cushion material. Then the package can be designed for resonances out of the range of frequencies most commonly encountered during transportation.

One problem of using the above procedure is that there are not enough handling data available. Wright Air Development Center is currently investigating package handling. And American Airlines has published its experience in handling and shipping. American found that loading and unloading is the greatest cause of damage. By sending an impact recording machine through its cargo operations, American found where improvements could be made and handling reduced. (Incidentally they found that 2.5 *g* was the maximum shock recorded during the air ride; up to 10 *g* impacts were recorded during ground handling.)

When further information is made available, the dynamic package test described above can become a useful tool in determining suitable packaging.

Types of Cushion Materials

The following types of cushion material are currently being used in packaging:

excelsior
corrugated fiberboard
paper, shredded
crepe paper wadding
moulded pulp
cotton wadding
felt, cellulose and hair
cellulose fiber and latex
animal hair and latex
glass fibers and resin
latex foam
blown sponge
molded shredded foam or sponge
plastic derivatives
shock mounts: rubber, spring, knitted wire

The quality of cushion material may be measured

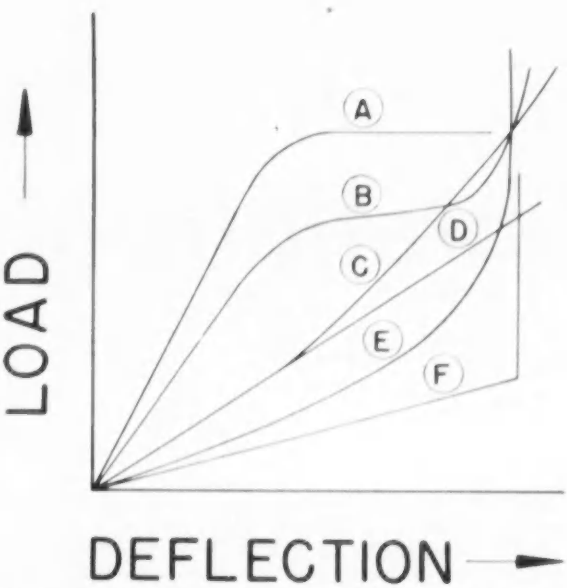


Fig. 1—Different suspension systems and cushionings react in different ways to loads. Best characteristics are nonlinear as shown in curves C and E.

by the amount of energy it can dissipate when subjected to shock. This is established by dynamic and static loading tests.

Geting standard figures is difficult because different cushion materials have radically different load deflections.

Fig. 1 illustrates several possible load-deflection characteristics. Curve A represents a method used to absorb maximum energy by a large deflection beyond a certain loading. However, the cushion material should have enough elasticity to return to shape after deflection. If mechanical shock mounts are used, care must be taken to prevent abrupt snubbing after the extended deflection. Additional resistance to loading, as illustrated by curve B, will correct this problem. Curves C and E show typical non-linear deflection characteristics most common to available cushion materials. Curve D is linear while curve F shows a typical deflection of a linear spring that bottoms out at a certain load.

Packaging materials should have nonlinear load-deflection characteristics to dissipate the maximum amount of energy within the cushion. This ability to resist loading progressively prevents large dynamic loads and resulting transient vibration.

Damping within a cushion material is necessary to prevent amplification of the suspended mass due to intermittent periodic disturbances. It also affects the maximum acceleration obtained under dynamic load. It reduces the natural frequency and the amount of rebound. For example, with 50% damping the tendency to rebound is reduced six times. Most of the time, however, the optimum amount of damping for the most effective over-all cushion characteristics is between 25 and 30%.

The simplest and most reliable way to measure damping is to record the wave form of the cushion material or the oscillatory transients set up after an initial shock loading. The damping ratios may be determined from the displacement ratio of successive peaks.

When designing a package and selecting cushion materials, the following information is necessary:

1. Maximum shock that the most fragile component of the item can stand without damage (fragility g).
2. Weight and size of the item.
3. Maximum shock expected to be encountered.

From this information, and with the help of a nomogram such as Container Laboratories, Inc. has developed for Bureau of Ordnance, Naval Ordnance Laboratory, it is possible to:

1. Estimate the minimum possible thickness of cushion material.
2. Determine the required thickness of a given cushion material.
3. Select a material which will provide a cushion of least thickness.
4. Provide the bearing area that results in a cushion of least thickness of a given material.

Shock Mounting Systems

The use of bulk cushion materials is not the only way to protect equipment from damage. For guided missiles, electronic control systems, jet engines, business machines, and computers shock mounting systems may be more economical.

A nonlinear type of deflection curve, as recommended for bulk cushion, is also necessary for shock mounting systems.

They should have a wide load tolerance without excessive deflection combined with good over-load capacity. Also, they should have good damping

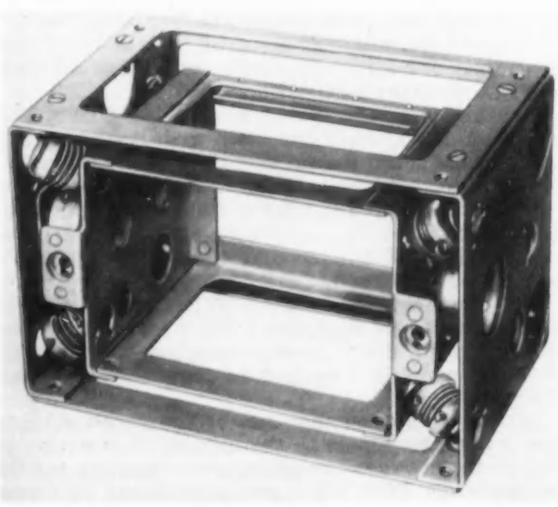


Fig. 2—Shock mounting elements are located at an angle to give effective center of gravity support.

efficiency unaffected by pressure, temperature, or humidity.

There are many different types of mounting designs using shock mounts. A base mounting combined with stabilizing resilient elements may be satisfactory for most cases. A modified system locates the mountings at the top and bottom of the equipment so that the diagonal lines of the top and bottom shock mounts intersect at the center of gravity of the equipment. Another system of this type uses mounting elements that are located at each end of the equipment and set at an angle to support the center of gravity. (See Fig. 2) Another type, shown in Fig. 3, has multi-direction shock protection by providing four specially designed, all-metal Robinson mounting beams located within and around a cylindrical container. Each beam is precompressed to the correct amount for this equipment to provide for the number of degrees of freedom. Stiffness in all planes may be varied to suit the packaging requirements. For long pieces of equipment the beam arrangement provides an excellent way of distributing the suspended load along the beams where convenient structural attachments may be made to the equipment. It is all-metal and uses resilient load-carrying cushions of knitted wire. It is particularly good for use in re-usable shipping containers.

The basic component of a flexible suspension (shock mounting) system is the resilient mounting, of which there are three fundamental types:

1. Plate-form mounting. This type consists of a metal plate and tube bonded to an elastomer. It is designed to have a low spring rate initially so that a wide range of disturbing frequencies can be isolated. A gradually increasing spring rate gives protection from shock loads. This is accomplished by having the static weight of the mounted mass carried by rubber-in-shear and shock loads by rubber-in-compression.

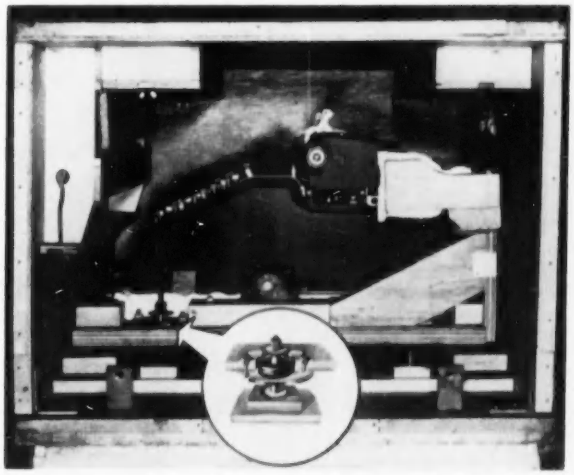


Fig. 4—Plate form mounting, shown in insert, is bolted to plywood base which is bolted to typewriter. It is located below center of gravity of the typewriter to induce rotation when the container is dropped on its side or end.

A typical installation of this type mounting in a shipping container is shown in Fig. 4. The equipment is clamped to a plywood base that is flexibly bolted to four plate-form mountings. Inner members are bolted to the containers. Spacers are used between the inner members and snubbing washers to provide greater travel and more protection before snubbing takes place. The mountings are located beneath the center of gravity of the machine to induce rotation of it on the shock mountings when the container is dropped on its side or end. Thus, the equipment is given greater lateral protection

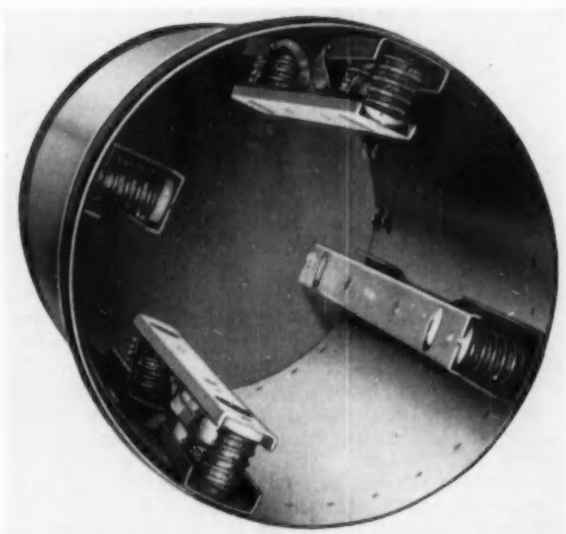


Fig. 3—Mounting beams provide multi-directional protection in a cylindrical container. Beams are precompressed to permit correct amount of freedom.

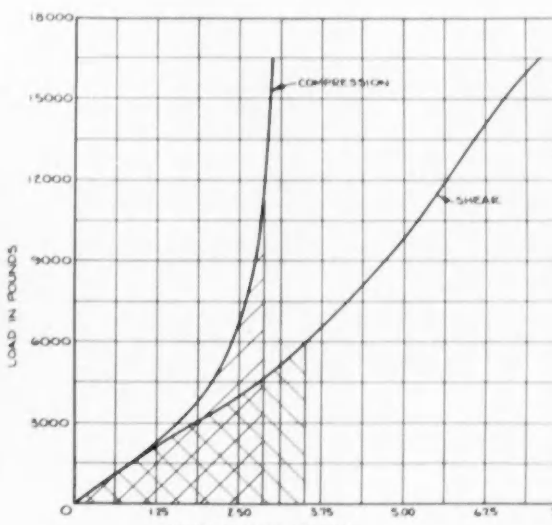


Fig. 5—Comparison of shear load-deflection curve with typical compression load deflection curve. Area beneath curves is directly proportional to the amount of energy which the mountings are capable of storing and dissipating. Shaded portions represent equal energies.

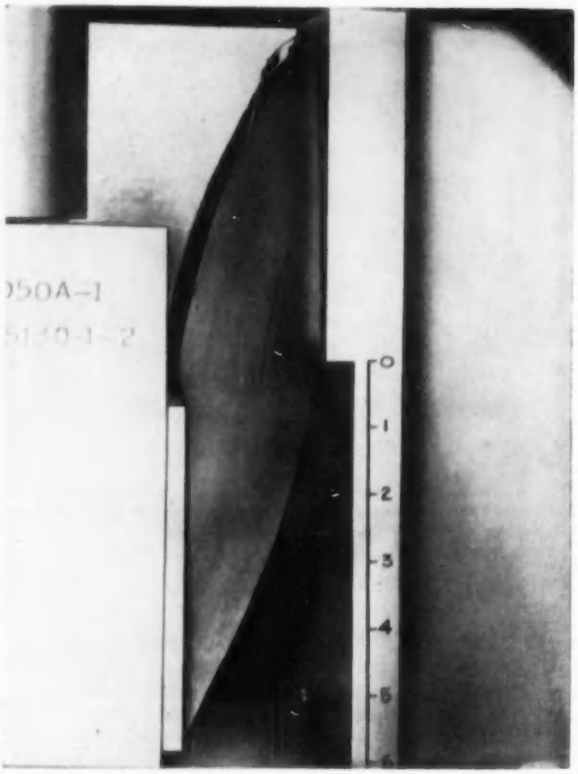


Fig. 6—A shear mounting in a deflected position. Its thin shape takes up very little space in the container.

than it would receive if it were limited to pure translation.

2. **Sheer sandwich mounting.** This type mounting consists of two metal plates bonded to an elastomer between them by dome nuts which are locked into the plates and protrude into the rubber section. The shear sandwich mounting has, essentially, a linear load-deflection curve and can be designed to produce a wide range of spring rates and deflections. Those available at the present time have spring rates varying from 20 lb/in. to 4200 lb/in., and some are capable of deflecting 12 in. linearly.

The linear load-deflection curve and large deflection of the shear sandwich mounting make it the ideal resilient member of a shipping container suspension. A suspension system consisting of shear mountings will permit only relatively small forces to act on the protected equipment, even though large amounts of energy are being exchanged between the equipment and the shear mountings. This can best be understood by comparing the parabolic load-deflection curve of a compression mounting with the linear load-deflection curve of a shear sandwich mounting (see Fig. 5).

Since the area beneath these curves is directly proportional to the amount of energy which the mountings are capable of storing and dissipating, the shear mounting with the straight line load-deflection curve will produce a much smaller force to absorb a given amount of energy than will the compression mounting with a parabolic load-deflection curve. Of course, the shear mounting must deflect farther than the compression mounting to do this.

The shaded portions beneath these curves represent equal energies. From these curves, it will be noted that the shear mounting deflects 3.5 in. to absorb this energy at a force of 6000 lb. The compression mounting, however, deflects 2.88 in. at a force of 11,250 lb. Equipment mounted alternately on the compression mountings and on the shear mountings represented by these curves and possessing the energy represented by the areas beneath these curves, would experience approximately twice the acceleration in being brought to rest on the compression mountings that it would on the shear mountings.

The shear mounting is very compact, requiring little space in the container. A shear mounting only $5 \times 5 \times 2\frac{1}{8}$ in. will produce a spring rate of 1700 lb/in. and deflection over 7 in. Fig. 6 shows this mounting in a deflected position. Its thin shape permits it to be located between the equipment and the container without increasing the width of the container over what is required anyway for clearance of the suspension system during motion caused by shock.

To protect equipment from shock in all directions, shear sandwich mountings are located beneath the center of gravity of the equipment. This induces rotation of the equipment when the container is shocked along the compression axis of the shear mountings. They sometimes are located at 45 deg angles to the side of the container to obtain the same deflection in all lateral directions. This type of design is illustrated in Fig. 7.

To obtain even more protection (for extreme conditions) the shear sandwiches can be compounded so that one set will shear horizontally and another will shear in the vertical plane. Fig. 8 illustrates this principle. Four shear mountings, with their shear planes vertical, support brackets to which are bolted four shear mountings with their shear planes horizontal. The cradle in which the equipment is clamped bolts to the top plates of the horizontal shear mountings. This gives excellent multi-directional protection without increasing the container size.

Occasionally it is necessary to mount equipment so that it receives protection at extremely low temperatures. An all-metal mounting is the only practical mounting which will operate at −65 F at the present time. There are two difficulties. First, to produce the necessary spring rate and deflection the mounting must be quite large, and it has to be positioned in the container at special locations to produce the required accelerations and stability. This increases the container size and cost. Second, it is difficult to make a multi-directional mounting out of a uni-directional spring.

The simplest mounting which meets this cold temperature requirement is actuated by a flexible cable which compresses a nest of coil springs between two plates. This method of loading the springs was chosen to permit the mountings to be located in the container in space that would not otherwise be used. Through the flexible cables, the springs can support the equipment at support points without being located there. Further, only one set of mountings is required. The flexible cables will

always compress the springs regardless of the direction in which the cable is pulled.

The all-metal mounting has a built-in friction damper to isolate vibrations and is quite readily installed in the container without requiring a special preloading device.

Whichever of the three mounting systems is used, it should offer as much protection in the rebound direction as in the direction of the initial shock. This is necessary because the containers themselves sometimes are as massive as the equipment they hold, so the rebound shock could be greater than the initial shock.

In making a container and suspension system for guided missiles the above principles apply, but the considerable amount of precision equipment in a missile and the extreme environmental conditions existing during transit and storage require additional protection.

Steel, aluminum, and glass reinforced plastic are acceptable materials for the container, although steel is probably the cheapest to produce, and provides rigidity, penetration resistance, blast resistance, and stamina under rough handling.

Any material that is used for the container must be protected from corrosion by a protective coating. Aluminum paint for a final coat will increase solar reflectivity, thereby reducing internal temperatures.

Internally, the container must be maintained at not more than 30% relative humidity by using a silica-gel desiccant. Corrosive solids, liquids, and gases should be kept to a minimum. This can be done by:

(1) Replacing cadmium and silver plate with other metals having better corrosion resistance.

(2) Applying a protective finish to the plate.

(3) Using a specific vapor absorbent such as activated charcoal or activated carbon.

(4) Purging the container periodically with dry air or nitrogen.

(5) Allowing the container to "breathe" through changes in ambient temperature.

The container must be easy to open and unload in a very short time. It must be designed so that it can be hoisted, skidded, lifted by fork truck, and stacked.

There are several different container designs that have been used for guided missile transport and storage. Each design has its own particular advantages and disadvantages.

The end-loading type container has a short-closure flange but requires an internal track for access to the component. This track is a serious liability in that if it is bent or damaged the contents can no longer be removed.

The external suspension-type container may be smaller than the top-opening container but when it is oscillating because of shock or vibration it becomes a hazard to personnel and equipment. Impacts directly on the suspended container are not isolated from the contents with external suspension methods.

Collapsible containers of various types are less bulky for return shipment and for knocked-down storage, but it becomes necessary to provide a coating on the missile to protect it from atmospheric conditions, since the container cannot readily be sealed. After shipment in the knocked-down condition containers are apt to be structurally weak and no longer usable. The collapsible containers are not suitable for floating ashore during amphibious operations.

The pallet-type container is preferred for unpressurized uses. However, it is unsuitable for pressure because of its large flat surfaces. It also requires added structure to support its suspension system.

The reusable, top-opening cylindrical steel container can be pressurized. It is simple to seal and easy to open. It gives complete protection from the environment, resists damage by penetration, protects the suspension mechanism, and can be floated on water.

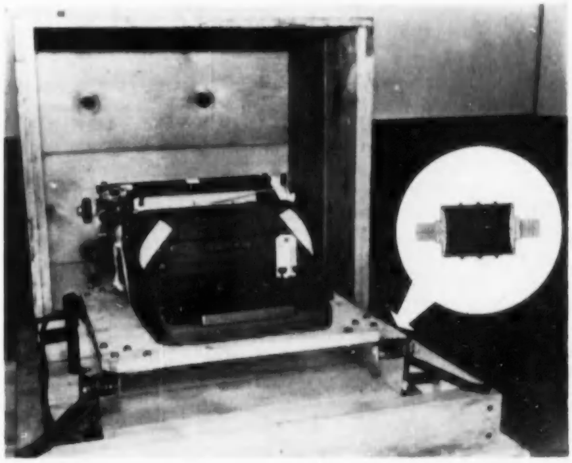


Fig. 7—Shear sandwich mountings are sometimes located at 45 deg angles to obtain the same deflection in all lateral directions. This typewriter was dropped along its three principle axes from a height of 60 in. with no damage.

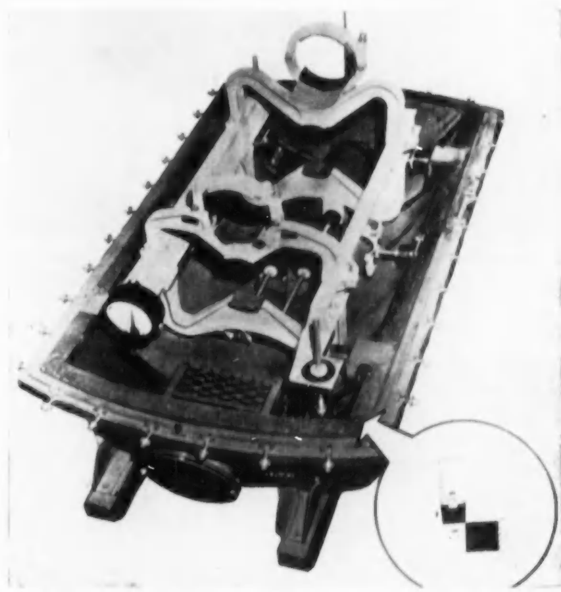


Fig. 8—Shear sandwich mountings can be compounded so that one set will shear horizontally, the other will shear in a vertical plane.

New Airborne Radar . . .

. . . detects thunderstorms and reveals intensity to permit pilot to avoid most turbulent areas. The unit has sweep ranges of 20, 50, and 150 miles.

Based on paper by **G. W. Church,** Bendix Aviation Corp.

THREE main operational uses are foreseen for the Bendix airborne weather radar. They are avoidance of thunderstorms, ground mapping, and beacon navigation (restricted to the X-band system at present, since no C-band ground racon stations exist). Field usage and the particular problems of the user will tell in time which is most important. The radar is not suited for detecting other aircraft.

The unit comprises an antenna, a transmitter-receiver, a synchronizer-power supply, an indicator with removable viewing hood, and a control panel. It is designed to operate at either X-band (9375 megacycles) or C-band (5400 megacycles) by the interchange of transmitter units and the replacement of portions of the antenna.

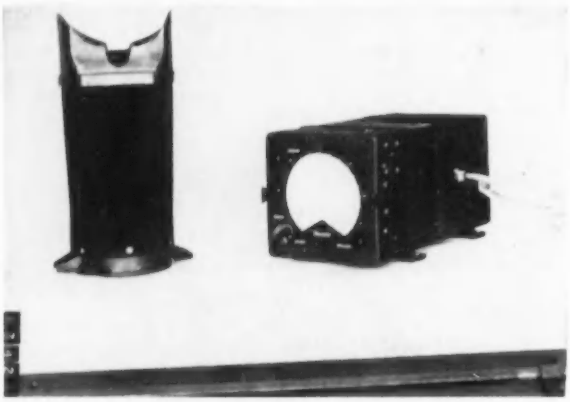


Fig. 1—Indicator of the Bendix airborne weather radar (right) with snap-on hood (left). The airplane's position is at the point of the inverted black V and there are three sweep ranges, giving a choice of 20, 50, and 150 miles.

Fig. 2—This is the way a series of thunderstorms looks on the indicator face of the airborne radar. In this instance the range is 50 miles and the range marks are spaced at intervals of 10 nautical miles.

The antenna is mounted in the nose of the aircraft under a radio-transparent but optically opaque dome fitted to the craft's contour. It scans the area in front of the plane continuously at a rate of 15 rpm. The transmitter-receiver and power supply units are mounted in the radio rack in accordance with standard airline practice of keeping electronic units in the pressurized area. The control unit is mounted in the cockpit convenient to the pilot and co-pilot. Both men have indicators mounted for easy observation in flight.

The face of the indicator (Fig. 1) is a long-persistence cathode ray tube which displays a plain map of the area being flown over. With the antenna pointed slightly down, coast lines, rivers, cities, mountains, and terrain features are shown in outline much as they might be drawn on a map. The airplane's position is at the point of the black inverted V. There are three sweep ranges on the indicator, giving a choice of a 20-, 50-, or 150-mile display as measured from the aircraft position to the top of the indicator. The range of display is reduced in the right and left directions by the geometry of the display and no picture is provided directly to the rear because the antenna cannot see through the aircraft. Range marks are provided at convenient intervals on the display, and there are radial marks at 30-deg increments of azimuth. The hood, snaps onto the indicator face to shield it from light if the cockpit is bright.

A series of thunderstorms on the indicator is shown in Fig. 2. The sweep range is 50 miles and range marks are at intervals of 10 nautical miles.

Isoecho circuits were incorporated in the radar to display a contour of storm intensity. In other words, the outer edge of the white area of a rain return defines the rainfall rate at which echoes are visible on the radar (near zero rain). The outer edge of the black hole in the center outlines the area within which the rainfall rate is greater than this by a pre-set amount, usually 12 to 16 db.

Where the rate of rainfall changes rapidly, maximum turbulence is usually found. Thus, where the white line between the no-rain area and the heavy-rain area is thinnest there is the maximum turbulence. Such a spot would be found in the cell shown at 25 miles and 22 deg left, upon entering it flying radially outward.

(Paper "A New Airborne Weather Radar" was presented at SAE Golden Anniversary Aeronautic Meeting, New York, April 18, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

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William B. Tilden ... Treasurer
R. Frank Wilson ... Secretary
Address of Secretary—3811 Durness St., Houston 25, Texas.

Twin City

Leslie W. Foster, Jr. ... Chairman
S. H. Knight ... Vice-Chairman
Albert L. Preston ... Treasurer
Donald D. Hornbeck ... Secretary
Address of Secretary—Casco Point, Rt. 1, Wayzata, Minn.

Virginia

Robert E. Porter ... Chairman
William H. Bingham ... Vice-Chairman
Charles W. Galloway ... Treasurer
William E. Bristow ... Secretary
Address of Secretary—Bristow Lines, Inc., 1605 Brook Rd., Richmond 20, Va.

Washington

Milo W. Snider ... Chairman
Harold D. Hoekstra ... Vice-Chairman
Bernard J. Vierling ... Vice-Chairman Aircraft
Charles H. McDonnell Vice-Chairman Engineering Materials
Ivison W. Rhodes, Jr. Vice-Chairman Transportation and Maintenance
Lewis C. Kibbee ... Treasurer
J. Chase Fielding ... Secretary
Address of Secretary—Fred H. Fielding Co., Inc., 1507 M St., N.W., Washington 5, D. C.

Western Michigan

Edwin C. Beck ... Chairman
Bernard E. Ricks ... Vice-Chairman
Earle J. Seward ... Vice-Chairman Grand Rapids
William B. Sketch ... Treasurer
Bine W. Rollin ... Secretary
Address of Secretary—Continental Motors Corp., Market St., Muskegon, Mich.

Wichita

R. H. Willey ... Chairman
Charles Y. Allison, Jr. Vice-Chairman
William W. Hurtt ... Treasurer
Robert G. McKay ... Secretary
Address of Secretary—242 N. Robin Rd., Rt. 8, Wichita, Kan.

OFFICERS of SAE Groups 1955-1956

Alberta

Charles H. Dawson ... Chairman
Norman Stone ... Vice-Chairman
Francis Price ... Treasurer
Wesley Moore ... Secretary
Address of Secretary—524 Crescent Rd., Calgary, Alberta, Can.

Colorado

John N. Gromer ... Chairman
William A. Nestlerode Vice-Chairman
Paul G. Anderson ... Treasurer
Richard F. Burris ... Secretary
Address of Secretary—6860 E. 73rd Pl., Derby, Colo.

Salt Lake

Wendell A. Hubbell ... Chairman
Harold E. LaBelle, Jr. Vice-Chairman
William P. Barnes Secretary-Treasurer
Address of Secretary—University of Utah, Room 308, Engineering Hall, Salt Lake City, 1, Utah.

Williamsport

Ewing W. Mueseler ... Chairman
Allen Weiss ... Vice-Chairman
Frank W. Riddell ... Treasurer
Paul Cervinsky ... Secretary
Address of Secretary—Montoursville, R.D. 2, Pa.

EMERY B. KEREKES, formerly with Loewy-Hydropress, Inc., has become associated with Allstates Engineering Co. as chief engineer. He will be responsible for all the technical aspects of the engineering, design, and development activities of the firm.

Kerekes plays an active role in the SAE National Production Activity Committee and helped plan the Aeronautic Production Forum in New York.

ROBERT SCHILLING and **MAURICE OLLEY** have been assigned a change in duties with Chevrolet Motors Division, General Motors Corp.

Schilling has been appointed director of research and development for the Chevrolet engineering department. He has been head of the General Motors Research Laboratories Engineering Mechanics department.

Olley has been re-assigned to special work dealing principally with suspensions for future cars and trucks. He has been serving on the SAE Passenger Car Activity Committee.

WAYNE H. WORTHINGTON has been inaugurated national President of the American Society of Agricultural Engineers. The occasion was the Society's 48th Annual Meeting held at the University of Illinois, Urbana, Ill.

Worthington was 1950 SAE Vice-President representing Tractor and Farm Machinery Activity. He is at present chairman of the Technical Committee.

WILLIAM C. HEATH, Solar Aircraft Co.'s executive engineer, marketing, was elected 1955–1956 chairman of the San Diego section, American Society of Mechanical Engineers. He is an SAE Councillor, was chairman of SAE San Diego Section 1950–1951, and was general chairman of the 1951 SAE National Aeronautic Meeting in Los Angeles.

RALPH J. ESCHBORN has been named chief engineer of Jack & Heintz, Inc. He joined the company in 1951 as staff engineer and became assistant chief engineer in 1953.

ALLEN M. ADAMS has been named president, Three Dimension Co., Division of Bell & Howell Co. He was appointed director of purchasing and material control in 1953 and continues to be responsible for these functions.

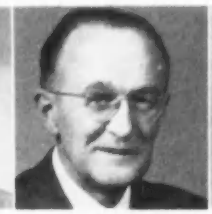
E. W. KETTERING, formerly chief engineer, has been named to the newly created post of director of research of Electro-Motive Division, General Motors Corp. The research department will concern itself with forward product planning, while the engineering department will carry on all engineering of products of Electro-Motive manufacture.

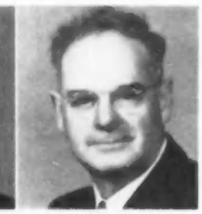
Kettering has been associated with projects contributing to the development of GM diesel locomotives since the beginning of his career. He has been chief engineer since 1948.

About SAE


Kerekes


Schilling


Olley


Worthington


Heath


Eschborn


Adams


Kettering

WALLACE E. CONNOLLY now has responsibility for component sales for Dalmo Victor Co. Sales Division. He is assistant director of sales in this newly developed Sales Division.

He had been associated with Lear-Romec Division of Lear, Inc. as East Coast contracts manager.

CRAIG MARKS is now associated with Ford Motor Co. as research engineer. He has been an instructor at California Institute of Technology.

HERBERT R. TAYLOR has joined Dryden Oil Co., Inc. as sales manager. He was previously assistant sales manager for Swan-Finch Oil Corp.

DR. NEWMAN A. HALL, educator and authority on the thermodynamic properties of materials, has been named to head the Graduate Division of New York University College of Engineering. He will serve as assistant dean and professor of mechanical engineering.

Previously Dr. Hall has been professor of mechanical engineering at the University of Minnesota.

DARREL D. DON CARLOS is now with Allison Division, GMC, as manager of development planning. He had been assistant chief engineer for the Buick-Oldsmobile-Pontiac Assembly Division.

ALEX PETROVSKY is now with Arma Division, American Bosch Arma Corp., as staff engineer—Production Engineering Department. Previously he served as staff engineer with Lockheed Missile Systems Division.

WILLIAM S. JOHNSTON, president of Johnston Cadillac, Inc. of Trenton, N. J., has been promoted from the rank of Colonel to Brigadier General in the United States Air Force Reserve.

General Johnston's promotion is in recognition of more than 30 years of active service in the Air Force Reserve. His decorations include the Legion of Merit and the Air Force Commendation Medal.

In his new rank, he has a Mobilization Day assignment of Assistant Chief of Staff for Operations, Headquarters, Tactical Air Command, Langley Air Force Base, Virginia.

Members


Klikunas


Vickers


Herman


Rowe


Rosenthal


Wright


Flaskamper


Abreu

EARL M. DOUGLAS is the new general manufacturing manager of the division recently established at Studebaker-Packard Corp. to coordinate all government and industrial products programs. The new division is known as the Government and Industrial Division.

Douglas has held the post of vice-president in charge of manufacturing.

OLIVER E. RODGERS has been named assistant general manager and chief engineer of the division. He has been chief engineer, Defense & Industrial Operations.

JOHN R. BARTHOLOMEW has been appointed chief engineer of the Axle Division of Eaton Mfg. Co. He has been assistant sales manager, with offices in Detroit, since 1948.

Bartholomew has been a member of SAE since 1922.

JOSEPH B. BIDWELL has been named head of the Engineering Mechanics Department, General Motors Research Laboratories Division. He has been serving as assistant head of mechanical development.

ARNOLD FINK has joined the staff of Esso Research and Engineering Co. and has been assigned to the Design Division.

LEONARD F. SWOYER, formerly Bristol, Conn. zone sales manager for New Departure Division, GMC, is now eastern regional sales manager.

EPHRAIM M. HOWARD is now in the Turbo Jet Facilities Group, Allison Division, GMC, as senior project engineer. His new duties involve design of test facilities for turbo jet engines and engine components. He formerly was senior experimental engineer with the same company.

HARRY L. SMITH, JR. has been elected a sales vice-president of Aluminum Co. of America. He will assume responsibility for directing the activities of Alcoa's New Kensington and Cleveland sales development divisions and the company's division of commercial research. He has been staff manager of product sales since 1952.

RICHARD V. KLIKUNAS is now connected with Buckeye Iron & Brass Works, Dayton, Ohio, as chief engineer. He had been director of engineering with Rensselaer Valve Co., Troy, N. Y.

HARRY F. VICKERS has been elected vice-chairman of the board of Vickers, Inc. The founder of Vickers, Inc., he is also president of Sperry Rand Corp., of which Vickers is a subsidiary.

KENNETH R. HERMAN was elected president of Vickers at the same time. He has been with Vickers since 1931, serving recently as vice-president and general manager. He is SAE Detroit Section chairman for 1955-1956.

JOHN E. ROWE has recently become vice-president of Ross Operating Valve Co. He joined Ross in 1947 as sales engineer and has served as plant manager for seven years.

W. C. ROSENTHAL, for 26 years a member of the International Harvester Farm Tractor Engineering Department, has retired from his position as of Sept. 1. He had been staff assistant to the manager of engineering.

He has been a member of SAE since 1917. He began his career with International Harvester as a designer in 1929.

DONALD G. WRIGHT now heads the appliance and furnace sales divisions of Perfection Industries, Inc. as general sales manager. He has been general manager of the Globe Stamping and Refrigeration Products Division of the Hupp Corp., whose central air conditioning business has been purchased by Perfection.

JOHN H. FLASKAMPER, former manager of fuel injection sales, American Bosch Division, American Bosch Arma Corp., has been advanced to the post of vice-president—Sales. He has been associated with American Bosch for 21 years.

FERNANDO DE MELLO ABREU is now manager of the Luboil and Fuel Oil Departments of the Companhia Portuguesa dos Petroleos BP (Portuguese organization of the British Petroleum Co., Ltd.). He previously served for 25 years with the Portuguese Co. of Atlantic Refining Co. as manager of Luboil Department.

ALFRED O. LUNING, formerly mechanical engineer with Lord Mfg. Co., Erie, Pa., has joined Westinghouse Electric Corp. Air Armament Division as associate engineer, assisting in mechanical design and coordination in development of radar equipment for national defense.

Erwin Frudden Retires

C. **ERWIN FRUDDEN** and Mrs. Frudden sailed August 17 on RMS Queen Elizabeth for a European vacation celebrating his retirement. They are shown here enjoying the sunshine for a moment in New York's Central Park the day before they embarked.

President of SAE in 1947, Frudden retired from Allis-Chalmers on July 31 after 25 years with the company and a whole career spent in tractor engineering. He had been a consulting engineer since 1944. Earlier he had been chief engineer of the tractor division. In both positions, he contributed widely to engineering understanding of farm needs in the way of tractors and attaching implements.

He will continue as a director of the Commercial Trust and Savings Bank of Charles City, Iowa—partly, he admits, because it gives him a good excuse for a monthly visit with his 2½-year-old grandchild, Leslie, daughter of the Fruddens' daughter Phyllis.

The Fruddens' itinerary on their European jaunt includes France, Italy, Switzerland, Germany, Denmark, and England. Most exciting prospect to them when they sailed was their expectation of a visit with their son, Bruce. He is employed abroad and will join them for part of their trip.

The senior Fruddens expect to return to this country on the Queen Elizabeth on Sept. 29. Frudden says his one regret is that—for the first time since he helped get them started on a regular basis in Milwaukee—he'll miss the SAE National Tractor Meeting, held this year Sept. 12-15.

P. G. ANDERSON has been appointed acting area automotive superintendent for Shell Oil Co.'s Denver Area. He joined Shell in 1954 as automotive engineer in that area. In his new position he replaces **HUGH CAUGHEY** who has been transferred to Shell Pipe Line Corp. in Houston, Texas. There Caughey will serve in the organization of Shell's new automotive department.

DONALD F. KEHN is now associated with Kudner Agency, Inc., New York City, as technical writer. He had been technical service engineer (Automotive) with Texas Co. in New York.

HENRY FORD II, president of Ford Motor Co., has been elected to the board of directors of General Electric Co. General Electric has increased its board membership from 16 to 18.

PAUL V. KEYSER, JR. has been elected a director of Socony Mobil Oil Co. He has been with the company since 1930, serving since 1951 as manager of domestic marketing.

MARSHALL J. WATERS is now in London, England with Shell Petroleum Co., Ltd., Oil Products Development Department. He is serving as technical consultant.

Previously he was manager, Products Application Department of Shell Oil Co. in San Francisco.

K. B. L. PERERA, managing director, K. B. L. Perera & Sons, Ltd., Ceylon, has been invested with the Order of the British Empire as a Queen's Birthday Honour on June 9, 1955.

ROBERT W. STODDARD, president of Wyman-Gordon Co., has been elected chairman of the board of Prex Corp., Chicago.

PAUL EDWARD TOMSHANY has moved to Highland Park, Mich. to serve as supervisor, Process and Maintenance Engineering, for the Ford Motor Co. Paint and Chemical Products Plants. He had been with the Ford Kansas City Aircraft Plant as supervisor, Manufacturing Engineering.

J. G. WATTERS is now associated with Iowa Mfg. Co. of Cedar Rapids as design engineer. He served in the same position with Herman Nelson Division, American Air Filter, Inc.

JOEL M. JACOBSON, vice-president and general manager of Aircraft Armaments, Inc., has been named executive vice-president of the company's new Canadian subsidiary, AA Engineering, Ltd. The new company will provide skilled armament system engineering to the Canadian Government modeled on the services presently being performed by Aircraft Armaments, Inc. for the United States Armed Services.

SAE PAST-PRESIDENT WILLIAM LITTLEWOOD, vice-president of American Airlines, Inc., has been elected vice-chairman of the Flight Safety Foundation's Industry Advisory Committee for 1955.

ELTON S. MOYER has been appointed assistant chief engineer of the Moraine Products Division of General Motors. He was formerly assistant chief engineer with Cadillac Tank Plant.

VERNON F. ELLIOTT has joined Curtis Hoover, Ltd. as an engineer concerned with cost estimating and design. He had been assistant plant and material handling engineer with Ford Motor Co. of Canada, Ltd.

JOSEPH KROLL is now located in Buenos Aires, Argentina as assistant chief engineer with Industrias Kaiser Argentina, S. A. He is serving with his company in the process of setting up an automotive industry in Argentina and will set up the main plant in Cordoba.

Formerly he was manager of the Experimental Engineering Division, Willys Motors, Inc.

ROBERT M. WILLIAMS has taken the position of deputy managing director of ARO, Inc., Tullahoma, Tenn. He was assistant to the president of Sverdrup & Parcel, Inc., St. Louis, Mo.

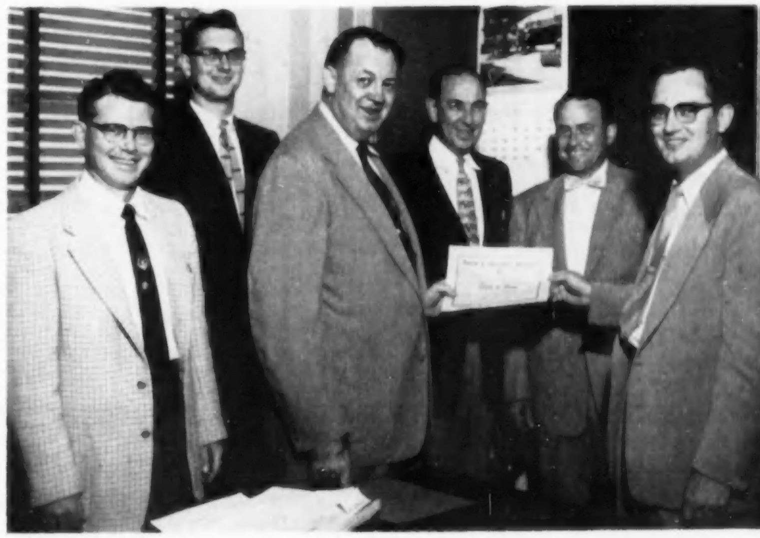
E. H. HEINEMANN, chief engineer, El Segundo Plant, Douglas Aircraft Co., Inc., has been awarded the Paul Tissandier Diploma by the Federation Aeronautique International at its general conference in Paris.

JOSEPH R. GILLETTE is now with the Ford Motor Co. Special Products Division as special assistant, merchandising and product planning. He had been special assistant, Public Relations, with Ford Lincoln-Mercury Division.

ROBERT G. REEVES has joined Lycoming Division, Avco Mfg. Corp., as development engineer. He was test engineer with John Reiner & Co., Long Island City.

ROBERT P. JENSEN, who has been Detroit field engineer for Kaiser Aluminum & National Lead Co., is now Baltimore branch manager for Kaiser Aluminum & Chemical Sales, Inc.

ROWLAND G. OONK has moved with his family to Dhahran, Saudi Arabia, to serve with the Arabian American Oil Co. as an automotive engineer in the Transportation Department. He had been serving with the company in New York as transportation engineer.



FRANK A. SUESS, director of the Sales Engineering Division of Continental Oil Co., has moved to Continental's offices at Houston, Texas. This move completes the transfer of staff personnel from the Ponca City offices to Houston.

Long an active member of Mid-Continent Section, Suess was presented a citation at his departure from Tulsa which reads: "The Governing Board of this Section takes pleasure in presenting this citation to Frank A. Suess, 'old timer' of the Section, in recognition of his constant loyalty and untiring effort in the Section."

COL. A. A. ARNHYM has been appointed Director of Information Services of the Air Research and Development Command, Baltimore, Md. This position entails responsibility for the public information, community relations, internal information, and historical activities of the headquarters and the Command.

Arnhym had been special assistant to the Commander of the Air Research and Development Command. He is retaining those responsibilities in addition to his new assignment.

WILLIAM C. SMOTHERS is now assistant chief Lincoln car engineer, Lincoln-Mercury Car Engineering Department, Ford Motor Co. He had been Lincoln development engineer.

RONALD W. GRIFFIN has moved to Ford Motor Co. Engineering Staff, Advanced Truck Engineering Department. There he is product engineering designer 'A'. He had been junior engineer with International Harvester Co.

WILLIAM A. PRICE has taken a position as development engineer with McDonnell Aircraft Corp., St. Louis, Mo. He was co-owner of the Price Garage in South Bend, Ind.

MILTON B. HAMMOND, JR., previously design engineer with Boeing Airplane Co., has joined North American Aviation, Inc. as research engineer.

J. WILLIAM SCHNABEL, formerly chief project engineer with Eastern Engineering Co., is now associated with Thompson Products, Inc. as consulting engineer.

LEWIS P. FAVORITE has been named manager of product sales with Aluminum Co. of America. Manager of Alcoa's New York district sales office since 1951, he assumes his new duties as coordinator of product sales activities in Pittsburgh.

LYNN R. PAUL has become works standards engineer in industrial engineering with Lincoln-Mercury Division, Ford Motor Co. He had been production process engineer for that division.

E. S. RUSSEY and **A. W. ROSE** have been elected vice-presidents of Borg-Warner Corp.

The vice-presidency conferred upon Russey recognizes his position as president and general manager of Warner Gear Division. He first joined Warner Gear in 1925 and became president in 1950. He is also a director of Borg-Warner Corp.

Rose resigned in 1953 as vice-president and assistant general manager of Warner Gear and simultaneously was named to represent Borg-Warner Corp.'s interests in financial and manufacturing circles in the Far West. His election as vice-president coincides with a current expansion of Borg-

CONTINUED ON PAGE 120

TECHNICAL COMMITTEE *Progress*

1955 SAE Technical Board

C. A. Chayne
Chairman
B. B. Bachman
D. P. Barnard
L. L. Bower
O. A. Brouer
G. E. Burks
H. E. Churchill
A. T. Colwell
F. W. Fink
E. G. Haven
R. F. Kohr
W. C. Lawrence
A. G. Loofbourrow
C. E. Mines
E. F. Norelius
Harold Nutt
A. E. Smith
B. G. Van Zee

Seidel Becomes Chairman of Committee E-21

DONALD SEIDEL (left) has been named chairman of Committee E-21, General Standards for Aircraft Engines, to succeed Robert F. Schwarzwalder (right). Seidel, who is with Westinghouse Electric, will assume the chairmanship at the committee's meeting in Dayton Oct. 11, 12, and 13.

Seidel

Schwarzwalder

Schwarzwalder, who is with Wright Aeronautical, has been chairman for the past two years. During that time, the committee has worked on 18 new projects and has organized five new panels. E-21 is a committee of the Aircraft Engine Division of the SAE Aeronautics Committee.

CRC Studies Diesel Fuel With Ignition Delay Bomb

THE results of a series of experiments studying the combustion characteristics of diesel fuel in a constant-volume bomb are presented in a CRC report "Combustion Characteristics—Ignition Delay Bomb, 1951-52." Tests were undertaken to learn more about the individual effects of engine and fuel parameters upon diesel starting and warm-up operations, in maintaining smooth combustion, and in controlling combustion deposits.

A constant-volume bomb was used and experimental techniques developed so as to provide a test tool that could be used for close control of the variables that affect combustion in the 650 to 1300 F temperature range.

Tests showed that the ignition delays of the aromatic portions of three different fuels were approximately two to four times that of the original fuel. Paraffin-napthene portions of the fuels gave ignition delays which averaged about 80% of the delays of the whole fuels. One of these paraffin-naphthene fractions, which had a high dicyclic content, showed approximately 20% increase in ignition delay as compared with the paraffin-naphthene fractions of the other two fuels.

A series of tests made on front-end volatility fuels showed no differences which could be attributed to the front-end volatility per se.

It was also found that the heavier fraction of residual fuel has comparable ignition delays to lighter diesel fuel cut from the same crude.

The report, CRC-279, has 45 pages, including six tables, 18 pages of charts and illustrations, and an appendix reviewing the early history of the project which had appeared in a previous report. CRC-279 is available from the SAE Special Publications Department. Price: $1.50 to members, $3.00 to nonmembers.

CRC Tests Motor Fuel For Light Aircraft

AN aircraft industry survey and series of tests to determine whether standard motor fuels could be used in light aircraft was conducted by CRC at the request of the Aeronautical Chamber of Commerce and the Air Transport Association. The results of this project are presented in the CRC report "In-

vestigation of Motor Grade Fuel as a Potential Fuel for Light Aircraft."

Automotive-type fuels were performance-tested by two major manufacturers, and cost and technical data on numerous possible fuels were distributed to the light engine and airframe industries soliciting their attitudes on the choice of fuel. The test program indicated that the use of motor fuel in light aircraft can adversely affect mixture control and power output, and can sometimes induce vapor lock. Results of the industry poll favored by a substantial margin the use of 80 octane number aviation grade fuel.

To use motor grade fuels in light aircraft, the fuel feed system, fuel metering system, intake manifold, exhaust valves, seats, and guides would have to be modified.

It was recommended that 80/87 octane fuel be used while 91/98 should be available for higher output engines.

Since the data and opinions contained in the report were obtained during the immediate postwar period, recently the conclusions were reviewed in the light of more recent data. They are considered to be still valid.

This report, CRC-282, has 23 pages, including six tables and two illustrations. It is available from SAE Special Publications Department. Price: $1.00 to members, $2.00 to nonmembers.

Lack of Funds Halts CRC Photography Project

UNFORESEEN problems in the early stages of a CRC project to investigate combustion phenomena by means of high-speed photography have used up allocated funds. So the program had to be stopped. Goal of the project was "to identify the phenomena occurring in a quartz-windowed engine combustion chamber and compare it with those occurring in a standard engine of the same basic type, and then to identify the physical phenomena as observed by high-speed photography with the associated engine-operating conditions."

Progress of the program up to abandonment is described in the CRC report "Investigation of Combustion Phenomena by Means of High-Speed Photography." Much information was obtained on high-speed photography techniques which will prove valuable to further research in this or related fields. There are no plans for starting up this activity again at this time.

The report, CRC-281, is available from the SAE Special Publications Department. Price: 50¢ to members, $1.00 to nonmembers.

Aero Materials Specs Reviewed by Industry

DRAFTS of thirty-two SAE Aeronautical Materials Specifications are currently being circulated to industry for comment and criticism by the SAE Aeronautical Materials Specifications Division.

Copies of all these specifications are available for review from the SAE Aeronautical Department, 29 West 39 Street, New York 18, N. Y.

The specifications under review are:

- AMS 3250B—Synthetic Rubber and Cork Composition, General Purpose Soft;
- AMS 3251B—Synthetic Rubber and Cork Composition, General Purpose Medium;
- AMS 3252B—Synthetic Rubber and Cork Composition, General Purpose Soft;
- AMS 3345—Silicone Rubber (1000 psi) (45–55);

AMS 3346—Silicone Rubber (1000 psi) (55–65);

- AMS 3347—Silicone Rubber (900 psi) (65–75);
- AMS 3357—Silicone Rubber, Lubricating Oil and Compression Set Resistant (65–75);
- AMS 3410D—Flux, Silver Brazing;
- AMS 4048B—Aluminum Alloy Sheet and Plate, Aluminum Alloy Clad 5.6 Zn-2.5-Mg-1.6Cu-0.25Cr (Alc. 75S-0);
- AMS 4049B—Aluminum Alloy Sheet and Plate, Aluminum Alloy Clad 5.6Zn-2.5Mg-1.6Cu-0.25Cr (Alc. 75S-T6);
- AMS 4441—Magnesium Alloy Castings, Sand, 3.5Ce-1Zr (EK31A-T6) Solution and Precipitation Treated;
- AMS 4442—Magnesium Alloy Castings, Sand, 3.3Ce-2.5Zn-0.8Zr (EZ33A-T5), Aged;
- AMS 4443—Magnesium Alloy Castings, Sand, 5Zn-0.7Zr (ZK51A-T5), Aged;
- AMS 5388B—Alloy Castings, Precision Investment, Corrosion and Heat Resistant Nickel Base — 16Cr-17Mo-4.5W-6Fe;
- AMS 5516E—Steel Sheet and Strip, Corrosion Resistant 18Cr-8Ni (SAE 30302), Cold Rolled-36,000 psi Yield;
- AMS 5573A—Steel Tubing, Seamless, Corrosion and Heat Resistant 17.5Cr-12.5Ni-2.3Mo (SAE 30316);
- AMS 5684B — Welding Electrodes, Coated, Alloy, Corrosion and Heat Resistant. Nickel Base—15Cr-9Fe-2(Cb + Ta);
- AMS 5690E—Steel Wire, Corrosion and Heat Resistant, 17Cr-12Ni-2.5Mo (SAE 30316);
- AMS 5727B—Steel, Corrosion and Heat Resistant, 16Cr-25Ni-6Mo;
- AMS 5728A—Steel, Corrosion and Heat Resistant, 16Cr-25Ni-6Mo Kellogg Electric Ingot Process;
- AMS 5735A—Steel, Corrosion and Heat Resistant, 15Cr-26Ni 1.3 Mo-1.9Ti-0.3V;
- AMS 6427A—Steel, 0.8Cr-1.8Ni-0.4 Mo-0.07V;
- AMS 7270C—Rings, Sealing, Synthetic Rubber, Fuel Resistant (65–75);
- AMS 7292—Labels, Aluminum Foil, Etched, Anodized and Dyed;
- AMS 72XX—Labels, Aluminum Foil, Etched and Enameled;
- AMS ——, Hard Coating Treatment of Aluminum Alloys;
- AMS ——, Magnesium Alloy Sheet, HK31-H24;
- AMS ——, Aluminum Sheet, Laminated, Edge Bonded;
- AMS ——, Aluminum Sheet, Laminated, Surface Bonded;
- AMS ——, Magnesium Alloy Castings, Sand, 3.5Ce-1Zr (EK31A-T5), Aged;
- AMS ——, Steel Tubing, Welded, Corrosion and Heat Resistant 18Cr-11Ni-(Cb + Ta) (SAE 30347), Thin Wall;
- AMS ——, Steel Tubing, Welded, Corrosion and Heat Resistant 18Cr-10Ni-Ti (SAE 3021), Thin Wall.

English-Speaking Countries Continue Standards Cooperation

AMERICAN - British - Canadian Conferences on Engineering Standards were held April 11 to 15 at the Engineering Societies Building in New York City. Representatives of the standards agencies, industry and the government services of the three countries, as well as observers from Australia, India, Pakistan, South Africa and New Zealand attended. The standards discussed were screw threads, acme threads, instrument threads, bolts and nuts, limits and fits, surface finish and drafting.

These conferences are a continuation of the co-operative effort established during World War II between the English-speaking countries. The last conference was held in June 1952 prior to the International Standards Organization Meeting at Columbia University.

Experiences with the Unified Screw Thread Standard adopted by the Declaration of Accord in 1948 were reviewed. Suggestions were made for improving the standards by tabulating the dimensions of the more generally used special diameter-pitch combinations. Designations for screw threads, general purpose internal thread minor diameter tolerances and gage tolerances, gage thread profile and gaging practice, were also discussed.

The Acme Thread Standards developed by ASA Sectional Committee Bl were reviewed. The British representatives reported surveys in their country indicated certain of the standard pitches specified for the general purpose Acme Screw Threads were not the most commonly required pitches for the standard nominal diameters concerned. An additional series of preferred diameters and pitches was proposed for consideration.

Instrument Screw Threads including miniature screw threads in the size range 0.25 mm to 1.4 mm diameter, commonly used in the horological industry and screw threads for microscope objectives and nose-pieces were discussed.

The proposed American draft on Miniature Screw Threads is generally in accord with the ISO recommendations. However, the British representatives felt it was premature to establish tolerances in view of the problems of gaging methods, gages and measuring equipment.

The proposed American draft on Microscope Objective Threads is in agreement with the Royal Microscope Society thread for diameter and pitch but shows a truncated form of Whitworth thread. The British representatives felt that full form Whitworth thread would be more desirable.

1955 SAE GOLDEN ANNIVERSARY NATIONAL MEETINGS . . .

September 12-15
Tractor Meeting and
Production Forum
Hotel Schroeder, Milwaukee
Wis.

October 11-15
Aeronautic Meeting
Aircraft Production Forum,
and Aircraft Engineering
Display
Hotel Statler, Los Angeles, Calif.

October 31-November 2
Transportation Meeting
The Chase, St. Louis, Mo.

November 2-4
Diesel Engine Meeting
The Chase, St. Louis, Mo.

November 9-10
Fuels and Lubricants Meeting
The Bellevue-Stratford
Philadelphia, Pennsylvania

1956 SAE National Meetings . . .

January 9-13
Annual Meeting
The Sheraton-Cadillac Hotel
and Hotel Statler, Detroit,
Michigan

March 6-8
Passenger Car, Body
and Materials Meeting
Hotel Statler
Detroit, Mich.

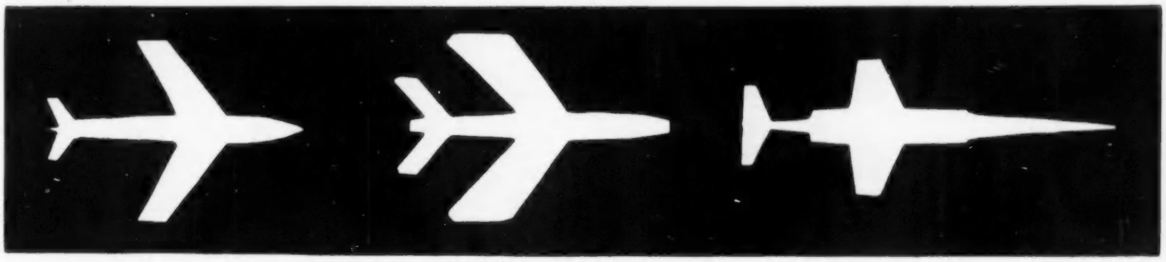
March 19-21
National Production Meeting
and Forum
Hotel Statler, Cleveland, Ohio

April 9-12
Aeronautic Meeting
Aeronautic Production Forum,
and Aircraft Engineering Display
Hotel Statler, New York, N. Y.

September 10-13
Tractor Meeting and
Production Forum
Hotel Schroeder, Milwaukee,
Wisconsin

June 3-8
Summer Meeting
Chalfonte-Haddon Hall
Atlantic City, New Jersey

August 6-8
West Coast Meeting
Mark Hopkins Hotel,
San Francisco, California



SAE Golden Anniversary

Aeronautic Meeting
Aircraft Production Forum and
Aircraft Engineering Display

October 11-12 Aircraft Production Forum

"Engineering and Manufacturing Look at Production Problems"

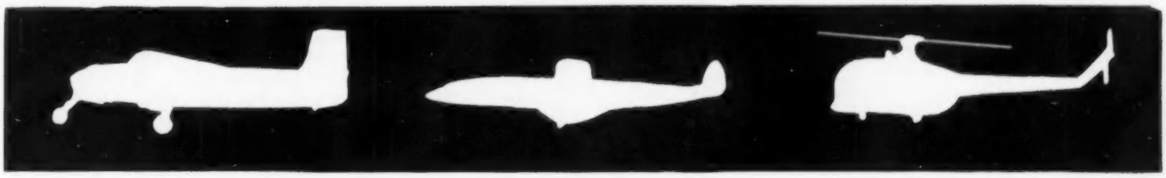
General Chairman: B. F. Raynes, Vice-President, Manufacturing,
Rohr Aircraft Corp.

October 13-14 Aeronautic Meeting and Engineering Display

General Chairman: E. H. Heinemann, Chief Engineer, El Segundo Division,
Douglas Aircraft Co., Inc.

Dinner-Dance
October 15 Hotel Statler

Hotel Statler **Los Angeles, Calif.** **October 11-15**



SAE Section Meetings . . .

Atlanta—September 12

Briarcliff Hotel. Dinner 7:00 p.m. Meeting 8:15 p.m. "Where Do We Go From Here?"—Charles Froesch, vice-president, engineering, Eastern Air Lines, New York, N. Y.

Canadian—September 12

Hamilton Golf & Country Club, Ancaster, Ont., Canada. Dinner 7:00 p.m. Golf in the afternoon. Special Features: A film titled "Taconite".

Central Illinois—September 26

Pere Marquette Hotel. Dinner 6:30 p.m. Meeting 7:45 p.m. "Packard Suspension"—M. A. Forrester, Packard Motor Car Co. Special Features: Presentation of Certificate to Past-Chairman Randall Roman.

Chicago, South Bend Division—September 19

La Salle Hotel, South Bend, Indiana. Dinner 6:30 p.m. Meeting 8:00 p.m. "1955 Indianapolis Racing Car Design" —R. T. Jackson, sales engineer, Eastern accounts, Perfect Circle Corp.

Cleveland—September 12

Chevrolet-Cleveland Division, GMC. Dinner 6:30 p.m. Meeting 7:30 p.m. "Suspensions for Commercial Vehicles" —Maurice Olley, special asst. to chief engineer in charge of Suspension Division, Chevrolet Motor Div., GMC., Detroit, Michigan. Special Features: There will be a plant tour through Chevrolet plant preceding dinner.

Colorado—September 12

Dinner 6:30 p.m. Meeting 8:00 p.m. "Power Steering in 1955"—W. K. Creson, consulting engineer, Ross Gear & Tool Co., Lafayette, Indiana.

Detroit—October 3

Large Auditorium & Banquet Hall, Rackham Educational Memorial. Passenger Car Activity Meeting. "The New Continental"—William Clay Ford, Ford Motor Co. Dinner Speaker to be announced.

Indiana—September 22

Marott Hotel, Indianapolis. Dinner 7:00 p.m. Meeting 8:00 p.m. "Automatic Machines for Automotive Plants" —W. Kent Mathias, Cincinnati Milling Machine Co., Cincinnati, Ohio. Special Features: Social Half Hour 6:30 p.m.

Metropolitan—September 15 and October 6

The Brass Rail Restaurant, Fifth Avenue between 43rd & 44th Sts. Cocktail Hour 5:30 p.m. Dinner 6:30 p.m. Meeting 7:45 p.m. "Testing Modern Aircraft"—Robert F. Wichser, chief research engineer, Republic Aviation Corp.

October 6—The Brass Rail Restaurant, Fifth Ave. between 43rd & 44th Sts. Cocktail Hour 5:30 p.m. Dinner 6:30 p.m. Meeting 7:45 p.m. "The Big Dividend from Research in the Greatest Power Industry of the World" —John M. Campbell, technical director, Research Labs. Division, GMC.

Mid-Michigan—October 10

Bancroft Hotel, Saginaw, Michigan. Dinner 6:30 p.m. Meeting 8:00 p.m. "Experimental Stress Analysis"—William T. Bean, Jr., consulting engineer.

New England—October 4

M.I.T. Faculty Club. Dinner 6:45 p.m. Meeting 8:00 p.m. "Motor Vehicle Exhaust Gases—Composition and Control"—George J. Nebel, research engineer, Fuels & Lub. Dept., General Motors Corp., G.M. Tech. Center, Warren, Mich.

Northern California—September 28

Engineers Club, San Francisco. Dinner 6:30 p.m. Meeting 8:00 p.m. "Relating to their experience with experimental items such as Paige & Paige Axle, Hall Scott Engines, etc."—Lewis H. Peterson, engineering superintendent, Pacific Intermountain Express, Denver, Colorado.

Oregon—September 15 and October 17

September 15—Pinky's, 20th & Lovejoy, N.W., Portland, Oregon. Dinner 6:30 p.m. Meeting 7:30 p.m. "Power Steering in 1955"—W. K. Creson, consulting engineer, Ross Gear & Tool Co., Lafayette, Indiana.

October 17—Pinky's. Dinner 6:30 p.m. Meeting 7:30 p.m. "Engine Temperature Why & How"—R. A. Weigel, president and general manager, Kysor Heater Co., Cadillac, Michigan.

Southern California—October 17

Rodger Young Auditorium. Dinner 6:30 p.m. Meeting 8:00 p.m. Passenger Car Activity.

Texas—September 16

Amon Carter Field. Display 6:00 p.m. Cocktails 7:00 p.m. Dinner 8:00 p.m. "Automotive Air Conditioning."

Texas Gulf Coast—October 7

A & M College of Texas.

Twin City—September 19 and October 12

September 19—Francis Drake Hotel, Minneapolis. Dinner 7:00 p.m. Meeting 8:00 p.m. "Thumbing Through an Engineer's Notebook"—1955 SAE President, C. G. A. Rosen, consulting engineer to president, Caterpillar Tractor Co., Peoria, Illinois.

October 12—St. Paul, Minnesota. Dinner 6:00 p.m. Meeting 3:30 p.m. Special Features: Field Trip through Ford Motor Co. Assembly Plant. Movie of "Mexican Road Race" and Trip through Ford's Glass Plant.

Williamsport—September 12 and October 3

September 12—Moose Club. Buffet Dinner 6:30 p.m. Meeting 8:00 p.m. Stag Affair with Films of VTO Aircraft and Atomic Submarine.

October 3—Moose Club. Dinner 6:30 p.m. Meeting 8:00 p.m. Experimental Engineers, Bell Helicopters, Texas.

Good Cooling Units Require Nine Features

Based on paper by

J. W. DUHN

Chrysler Corp

IF air conditioners are to do a satisfactory job of car cooling they must meet the following nine requirements:

1. Be able to cool a car rapidly and keep it cool regardless of speed or outside conditions.
2. Provide a wide-range control, simple and easy to operate.
3. Have big outlets and low air velocity to provide even, draft-free air distribution.
4. Filter and dehumidify to remove all dust, pollen, and moisture.
5. Admit fresh air to avoid staleness.
6. Operate quietly.
7. Avoid fogging windows which might cause condensation and staining of interior fabrics.
8. Not interfere with normal engine cooling.
9. Have neat, built-in appearance, with no extraneous hardware.

(Paper "Some Cold Facts on Car Air Conditioning" was presented at SAE Texas Gulf Coast Section, Houston, March 11, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers).

New Cutting Tools Aid Aircraft Production

Based on report by

L. W. WAITT

General Electric Co

AIRCRAFT production's unique problems, such as limited output and hard metals, have given rise to the development of new (and revision of old) metal-cutting methods.

Hydro Spinning, a recent development, displaces or extrudes metal over a revolving cylindrical mandrel by applying very high pressures to a roll which actually reduces the thickness of a sheet blank or extrudes a developed blank into an elongated formed part. When the process is applied to a straight tapered mandrel using a flat blank, the thickness of the blank required will be found by dividing the thickness of the tapered wall section by the line of the angle of the mandrel.

In the field of jet engine work, the Elox electrical discharge machine makes possible the piercing of intricate shapes in their unsupported fabrications. Since it differs from a punch and die in requiring no support, thin multiple thicknesses can be cut through. As a result, many very light, intricate, fabricated parts are being designed which were impossible, hitherto, to produce.

Another application is in grinding where the cylindrical electrode replaces the abrasive grinding wheel. Tests indicate that the cost (including labor) of sharpening Elox electrode wheels is considerably less than that of sharpening carbide tools. The carbide at the cutting edge is free from stress with this method and that is thought to ex-

plain the increase in cutting tool life—which is as much as 300%.

Chemical milling is still another development. This method controls the removal of material by use of acid, and masking areas not to be affected. The removal can be controlled at from 0.001 to 0.004 in. per min, and when solutions are well controlled, dimensional accuracy can be held within limits of 0.001 in. Stainless steel, titanium, and aluminum have all been milled successfully by this process.

The economical manufacture of many gas turbine engine parts has been made possible by the use of broaching. Cutting speed is limited to approximately 300 fpm because of the need to stop and reverse the ram. Titanium requires the very slow speed of 10 to 12 fpm for success, with broach life about one-third that expected in cutting stainless steel. Finish is excellent.

Chain broaches are used successfully where the production justifies additional tooling costs. Cutting speeds are greater than with the conventional reciprocating type of machine, while broach cutter life is doubled or tripled. Production in parts per hour can be expected to be two to three times greater with the chain broach. However, it will handle only parts that can be carried on the chain, and this limits the size of piece that can be worked.

(This article is based on the secretary's report of Panel on The Effects of a Cost Reduction Program on Tooling held as part of the Aeronautic Production Forum at the SAE Golden Anniversary Aeronautic Meeting, New York, April 21, 1955. It is available in full together with six other reports as SP-311, from SAE Special Publications Department. Price: $1.50 to members, $3.00 to nonmembers.)

Experts on the Cost Reduction Panel were . . .

Panel Leader

C. P. Urbon,
General Electric Co.

Panel Secretary

L. W. Waitt,
General Electric Co.

O. Bonnafe
La Pointe Machine Tool Co.

C. P. Brooks
Austenal Co.

E. Dachor
Thompson Products, Inc.

G. W. Periman
North American Aviation Corp.

R. Winblad
Cincinnati Milling Machine Co.

Inspection Systems Can Be Made Simpler

Based on paper by

PAUL E. ALLEN
Beech Aircraft Corp.

TO keep product cost reasonably low and still retain assurance of product conformity, control, reliability, and service is a difficult problem. It gets tougher with the growing complexity of aircraft equipment.

In our experience, adding to inspection forces does not keep malfunctioning equipment from reaching flight

test areas. Multiple inspection merely boosts costs and worsens the problem of getting qualified manpower.

When analysis revealed that a large percentage of engineering drawing dimensions are primarily to aid tool and jig design and intergroup coordination and make no contribution to the end product's quality or reliability, engineering was asked to try classifying all drawing dimensions into four categories. Then preliminary experiments were made.

It was found that in one group of machined parts 43% of dimensions were considered unimportant by engineering as long as the part made a satisfactory assembly; 62% of dimensions of a group of sheet metal parts were similarly classified. This meant a saving in inspection man-hours and reduction in number of minor reworks. Furthermore, cutting the number of items to be checked by one-half meant reducing the chances of inspection error by a like amount.

An example will show how this basic engineering classification works. A machine part has 18—or 50%—of its dimensions which need not be controlled. A time study on one order of 22 such parts indicates that 215 min are required for one inspector to check all dimensions, but only 120 min to check dimensions specified as necessary. That's a saving of 44%. The number of individual checks was cut from 682 to 279 and, given a 1.0% human error rate, errors would be reduced from 6.8 to 2.8 for the lot of parts. The inspector's production is raised and accuracy improved.

A more complex case is represented by a windshield defogger assembly. Here there are 95 pieces in a sequence of parts formed from nine dies. If processed through 100% inspection of all drawing dimensions there would be an estimated 10 rejection tags written at a cost of $150. And if the tools were reworked to fabricate parts to meet all basic drawing requirements, the rework cost would be about $1200. Exact conforming parts would be of no better quality than the ones already produced; hence, any drawing changes, salvage expense, or tool rework would be a waste of time, money, and effort. The time required to classify the characteristics is insignificant compared to the salvage time on the first order of parts produced.

This inspection system meets the requirements of MIL-Q-5923B and gives a product control that has passed both Civil Aeronautics Administration and Air Force surveillance checks and surveys. It also eliminates the uncontrollable old-fashioned type of so-called horse-sense inspection which has led many a manufacturer into some serious trouble.

(Paper "Simplified Inspection Techniques" was presented at SAE Wichita Section, Nov. 13, 1954. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Cost Control Success Depends on Five Rules

Based on report by

L. L. BLAKE

Bendix Products Div., Bendix Aviation Corp.

A COST control system is basically a management tool. And it must be tailored to meet the specific needs of a particular company. There are five rules essential for its success.

1. Responsibility for cost should be assigned clearly and logically to one individual in a department, function, or center. That person should be held accountable for only those costs over which he has control authority.

2. Actual costs should be measured against standards which represent high-level but attainable performance. Standards should be developed through sound analysis, preferably through industrial engineering techniques.

3. Those responsible for cost control should be supplied with reliable and current information. The data should be simple yet complete and accurate, and in sufficient detail to give adequate identification. Actual cost and comparative standard must contain the same ingredients.

4. Performance reports should be issued on a current basis while events are fresh in the supervisor's memory.

5. Follow-up should be organized to insure positive corrective action. If control data are not used actively to improve cost performance and correct unfavorable conditions when they exist, all the work and expense of developing standards and reporting performance is wasted.

(The report on which this article is based is available in full in multilith form together with reports of the seven other panel sessions of the 1955 SAE National Production Forum. This publication, SP-310, is available from the SAE Special Publications Department. Price: $1.50 to members; $3.00 to nonmembers.)

Cost Control Panel . . .

Panel Leader

E. O. Wirth,
Bendix Products Division, Bendix Aviation Corp.

Panel Secretary

L. L. Blake,
Bendix Products Division, Bendix Aviation Corp.

B. F. Armbruster, The Oliver Corp.

Ed Martin, General Electric Co.

J. W. Pocock, Booz, Allen & Hamilton

John Roller, R. K. LeBond Machine Tool Co.

H. A. Williams, Eaton Mfg. Co.

You build the right cab for your truck...

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Your heater, like the cab for your truck, should be designed and built to meet your own truck requirements.

The heater Evans engineers design and custom-build for you will meet *all* your truck requirements. It will fit *right*—for quick, easy installation. It will deliver all the BTU's needed for maximum driver safety and comfort—under *any* weather conditions.

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Complete truck and bus systems . . . built right for the job!

EVANS

Versatility Is Key To Power Shovel Design

Based on paper by

Q. J. WINSOR

The Thew Shovel Co.

THE problem in commercial-size power shovels is to provide the powerplant and the design strength for all the front end attachments the manufacturer wishes to provide. A power shovel may be converted into an efficient dragline, clamshell, pile driver, crane, or hoe.

Power has to produce a bail speed and pull and crowd speed and push capable of pushing and pulling a certain size dipper through the material to be dug. This comes down to force per inch of dipper cutting width. Methods of preventing stall in hard digging include fluid couplings, torque converters, and overload clutch releases.

Power also has to provide the swing acceleration and deceleration, and travel ability for walking around or climbing a rated grade.

(Paper "Power Shovels for Excavation and Materials Handling" was presented at SAE Earthmoving Industry Conference, Peoria, Ill., April 13, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Valve Lifter Noise Can Be Abated

Based on paper by

C. K. PARKER

and

M. W. SAVAGE

California Research Corp.

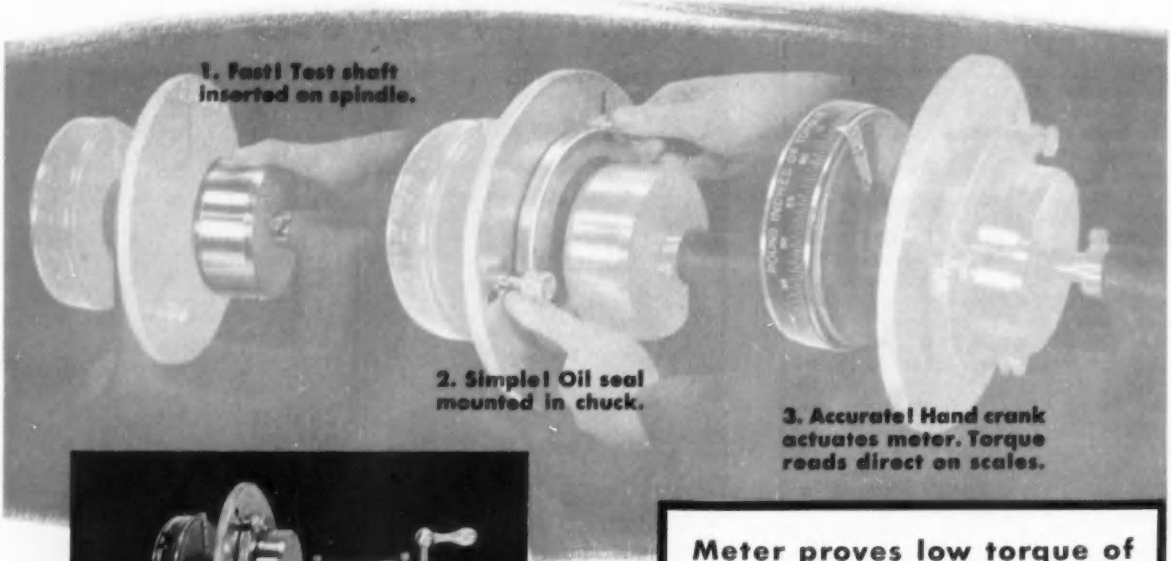
MOST cases of hydraulic valve lifter noise follow a typical pattern. The lifters function properly and give noise-free operation for the first 10,000 to 15,000 miles. During the next 5000 to 10,000 miles there may be occasional noise but it will not be frequent or persistent enough to annoy the car owner. When a mileage of 25,000 to 30,000 is reached, the noise becomes more frequent and the lifters need changing.

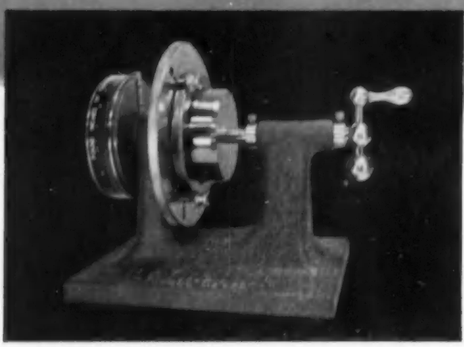
Measurement of lifters shows that during this last period the most significant change was an increase in leakdown rate—that is, the measure of the clearance between the plunger and the body of the lifter. Wear of plunger and body is then the major cause of noise.

In many cases, the ball travel was greater than the manufacturer's speci-

Now! MEASURE OIL SEAL TORQUE

quickly, accurately, by hand!





New NMB Portable Torque Meter

For years, industry has needed a fast, accurate means of measuring oil seal torque for standards determination or during quality control and parts inspection.

The new NMB Portable Torque Meter offers laboratory accuracy for such measurements, yet is rugged, fool-proof, and easily used by production line or receiving department personnel. A minimum of operator training is required; readings are instantaneous and direct.

Two models are offered. Model 615 reads torque to 15 lb. in. on seals from $\frac{3}{4}''$ to $6''$ diameter; Model 845 reads torque to 60 lb. in. on seals from $2\frac{3}{4}''$ to $8''$ diameter. Both have a simple, self-centering chuck which accurately positions the seal, and employ standard tapered shafts (special to order). The meters are lightweight (about 27 lbs.) and quality built for years of dependable service. A carrying case is available.

Meter proves low torque of NATIONAL OIL SEALS

National Torque Meters give quick, convincing proof that National Oil Seals have lowest torque.

National Micro-Torc Leather Oil Seals have appreciably less torque than conventional leather seals. National Syntech® synthetic rubber seals also test materially less torque than ordinary synthetics.

Micro-Torc seals achieve their remarkably low torque from a high lubricity coating applied to the surface of the sealing member. (Inside of sealing member retains natural porosity and stores oil for semi-starved operation.)

Syntech Oil Seals have the famous National "minimum-contact" sealing lip which, combined with precision tensioning, insures the lowest torque consistent with effective sealing.

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DETROIT, MICH.	726 Lothrop Avenue, TRinity 1-6363
DOWNEY (Los Angeles Co.), CALIF.	11634 Patten Rd., TOpaz 2-8166
INDIANAPOLIS, INDIANA	2802 North Delaware St., WAlnut 3-1535
MILWAUKEE, WIS.	647 West Virginia Street, BRoadway 1-3234
NEWARK, N. J.	Suite 814, 1180 Raymond Blvd., MItchell 2-7586
REDWOOD CITY, CALIF.	Broadway and National, EMerson 6-3861
WICHITA, KANSAS	519 South Broadway, WIchita 2-6971

fication. And it was also observed that material removed from the interior of the lifters contained numerous particles large enough to wedge between the plunger and body of the lifter and impede the action of the plunger. This combination of wear of the lifter and presence of solid particles occurred in almost all cases of lifter noise.

The car owner can do a number of things to extend hydraulic valve lifter life and minimize noise. The most important of these is to maintain a proper oil drain interval to remove contaminants. Use of adequately compounded lubricants and reasonable oil drain periods will maintain effective oil compounding in the crankcase to combat the corrosive wear of the lifter plunger. Oil filtration and adequate filter changes also remove material from the oil which can cause wear and noise. An oil change will often make the noise disappear. Finally, the use of a 180 F thermostat in the cooling system will extend lifter life by maintaining the effectiveness of lube oil additive, increasing filterability of the oil, reducing corrosive wear of the plunger, and reducing the amount of sludge collected in the oil.

(Paper, "Lubrication of Hydraulic Valve Lifters" was presented at SAE Seminar on Fuels and Lubricants, Los Angeles, April 5, 1955. It is available together with five other papers as SP-139 from SAE Special Publications Department. Price: $1.75 to members, $3.50 to nonmembers.)

Modern Road Builders Seek Better Equipment

Based on paper by

NELLO L. TEER, JR.

Nello L. Teer Co.

TURNPIKE construction, involving the moving of vast quantities of material under highly competitive conditions, is making increased demands on earth-moving equipment.

Reliability is the first requisite. The contractor must have machines that will do the job with a minimum of maintenance and down-time. If he can count on his equipment he can plan work to take advantage of the weather and complete his work with a minimum of capital equipment and at minimum cost.

Next, he wants reserve—ample power to meet adverse conditions, and ample strength to sustain shock. Too much equipment is under-powered and too weak to maintain production. There are trucks that have to shift down to low-low on normal construction grades or wet and muddy haul roads; power shovels that can't hoist the bucket up through the bank without stopping to let the engine "rev" up; axles, clutches, and transmissions that last weeks instead of years; tires that blow out due to inadequate carrying capacity. We are interested in final cost, not first cost. Ample reserve means lower maintenance, less down-time, and lower unit cost.

Finally, the contractor wants capacity. Everything being equal, larger units are more desirable. Use of 4 to 5-cu yd shovels and 20 to 25-yd trucks, with supporting equipment to match, is on the increase. But with increase in weight and dimensions comes a transportation problem. There is needed an over-the-road truck tractor and low boy trailer with enough aluminum or alloy components to help overcome gross weight limitations now in effect on highways. In many States, existing crawler tractors, without attachments, can be moved only by securing permits at so much a mile after

expensive telephone requests and great delays, and perhaps only by going 100 miles out of the way to follow designated routes.

Whatever the size of equipment it must be good and it should be field tested by contractors before marketing. Test farms don't reflect contractor usage. (Paper "Organizing and Operating a Turnpike Job" was presented at SAE Central Illinois Earthmoving Industry Conference, Peoria, Ill., April 14, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Oil Industry Acts To Curb Smog in L. A.

Based on paper by

CARL V. KANTER

Los Angeles County Air Pollution Control District

EVERY day about 1150 tons of hydrocarbons find their way into Los Angeles atmosphere from automobile exhausts. Production, refining, and marketing of petroleum products accounts for an additional 300 tons. At the moment there is no solution to smog caused by automobile exhaust, but something can be done to stop losses from handling of petroleum products, and the oil industry is doing it.

In April, 1953, a rule was passed requiring all gasoline storage tanks of over 40,000-gal capacity to have vapor recovery facilities or a floating roof, since losses from evaporation were found to be a substantial factor in contributing hydrocarbons to the atmosphere. The rule has been complied with 100%, and storage loss has been reduced to a minimum.

Transfer of gasoline to tank trucks and tank cars, to service stations, and to cars, was responsible for losses to the atmosphere of some 50 tons daily. Methods of reducing losses have been studied and several installations have been made at bulk stations.

At refineries open oil-water separators were responsible for at least 100 tons daily of hydrocarbon evaporation. Installation of closed systems or floating roofs has stopped most of this. Where blowdown stacks have emitted large quantities of hydrocarbons, facilities have been installed to capture the vapors and return them for processing.

Finally, many steam ejector systems on vacuum distillation equipment have been modified so that all otherwise non-combustible components are burned in furnaces or put into the fuel gas systems. (Paper "Sources of Air Pollution in the Los Angeles Area" was presented at the SAE Seminar on Fuels and Lubricants, Los Angeles, April 6, 1955. It is available together with five other papers as SP-139 from SAE Special Publications Department. Price: $1.75 to members, $3.50 to nonmembers.)

Oil Test Should Tell Detergency Activity

Based on paper by

RALPH FABER

Faber Laboratories

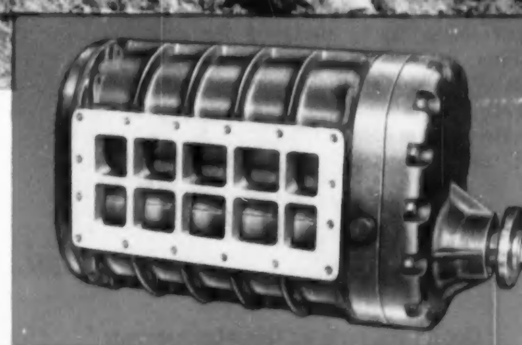
WHEN all contaminating particles in an oil are kept finely divided and suspended in microsize, the purpose of adding detergent to the oil has been accomplished. It does not matter what kind of detergent is used or how much of it was put in originally. If all particles are microsize the detergency activity is 100%.

As a result of making thousands of Faber detergency activity tests, which consist of separating contaminating particles from oxidized oil and oxidized fuel into groups by size and weight through the use of selective coagulants, and from an equally large number of tests for remaining detergent compound, several conclusions have been reached. These are:

A test for amount of remaining detergent compound is not the criterion upon which any reliability can be placed. This would be true even if methods were developed which could make the determination accurately and economically for all oils or for all detergent compounds.

Each engine is unique, regardless of make, and each may have different rates of creating oxidized oil and oxidized fuel contamination. An oil with little detergent but with a light contamination burden can have a 100% detergency activity. An oil with a large amount of detergent can have a very low detergency activity if a heavy contamination burden is placed upon it. The important thing is not to know how much detergent compound remains, but to know what the engine is doing to the detergent. (Paper "The Faber Test for Per Cent of Detergency Activity of Oil" was presented at SAE Seminar on Fuels and Lubricants, Los Angeles, April 4, 1955. It is available together with five other papers as SP-139 from SAE Special Publications Department. Price: $1.75 to members, $3.50 to nonmembers.)

GIANT HD-21 CRAWLER DOZES with Supercharged Power!



ALLIS-CHALMERS STANDARDIZES ON MIEHLE-DEXTER SUPERCHARGERS FOR THEIR DIESEL ENGINES

Like so many leading Diesel engine manufacturers, Allis-Chalmers specifies Miehle-Dexter Superchargers for their engines.

And that's one of the reasons this new giant torque converter driven crawler tractor—the Allis-Chalmers Model HD-21—claims to be one of the most powerful crawlers . . . anywhere! Rough and rugged dozing needs power aplenty. And here's *extra* speed at any load, *extra* pushing or pulling power at any speed, wider speed range and more range overlap—thanks to Miehle-Dexter Supercharged Power.

You, too, can boost the horsepower of your engines by 50% or more . . . and decrease weight per horsepower, too. Miehle-Dexter engineers can specify the exact size and type M-D supercharger for your engines. Standard M-D models available for applications to 750 hp. Write for bulletin.

FOUR FEATURES PROVE MIEHLE-DEXTER SUPERCHARGERS BETTER ON THE JOB

- Patented end plate seals eliminate metal-to-metal contact, assure longer life, help achieve fuel savings.
- Rotor wear strips eliminate contact between rotors and housing, assure longer wear.
- Lightweight aluminum rotors and aluminum case boost engine horsepower, do not add weight, safely allow high speeds.
- All parts are standardized, allowing easy field service and interchangeability.

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Division of the Dexter Folder Company

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Develops Earthmovers For War of Future

Based on paper by

BRIG.-GEN. HOWARD KER

U. S. Army

THE Transair tractor, now under test, exemplifies the Army's concept of a multipurpose construction equipment item of the type needed in future conflict. It incorporates air transportability for the assault phase of an airhead operation with an airlift weight of only 11,000 lb but a working weight of 50,000 lb by using local ballast. It is a prime mover, a bulldozer, and an air compressor. And it has a cross-country speed of 35 mph.

Among other specials in the development stage are an airborne shovel and a huge earth augur. There is also a rubber-tired ditcher which digs foxholes at the rate of one a minute and trenches at the rate of 20 fpm. It is known as the Infantryman's Dream.

These items and many others will be the tools equiping the military engineer to meet the increased demand for construction which modern war will create. (Paper "Construction Power in Military Operations" was presented at SAE Central Illinois Earthmoving Industry Conference, Peoria, Ill., April 14, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Easy Maintenance Is Earthmover "Must"

Based on paper by

FRANK SKIDMORE

Associated Equipment Distributors

INITIAL cost is not the prime cost of earthmoving machinery. The salesman can overcome the handicap of price if he knows that the unit he is selling will deliver the promised performance and justify the cost.

These are some of the things the salesman wants in the product he is to sell:

Equipment that has been thoroughly tested, preferably in the field. New developments and new machines should be job-tested by the producer.

Units that are ready for operation without the need to tighten nuts or undergo a break-in period.

Modernity in style—eye appeal and good lines.

A complete range of sizes so that he can fit the machine to the job.

Simplicity in mechanical design to simplify service and maintenance.

Assemblies and components so de-

signed that they will not only stay together once installed, but can be taken apart easily. Skilled help is not always available, particularly in the field remote from machine shop and special tools. Time saved in disassembly and reassembly means time to put on productive work.

(Paper "What the Salesman Wants to Sell in Earthmoving Machinery" was presented at SAE Central Illinois Earthmoving Industry Conference, Peoria, April 14, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Wanted! Chassis Lube With Longer Life

Based on paper by

L. J. KEHOE, JR.
General Motors Corp.

CHASSIS lubricants have failed to meet supposed greasing schedules ever since independent front suspensions came on the scene, or for the past 25 years.

There seems to be no generally distributed grease capable of really adequate lubrication of bushings, king pins, or metallic joints. In the past, we used threaded bushings on the rear of a car. These still have many advantages, such as lateral rigidity and good life (when properly lubricated). We now use rubber bushings on rear suspensions and they have several disadvantages. They add to the rate of the suspension, have shorter life than a threaded shackle, and complicate getting sufficient lateral rigidity. But they still have one advantage: they give a consistent pattern of performance.

The rear suspension with rubber bushings and inner liners in the springs has the virtue of consistency. But the same cannot be said for front suspensions. A keen observer, familiar with a car's handling after lubrication, can feel a change for the worse after 10 miles of driving on a wet day, or 100 miles on a dry one. The average driver should be able to sense the difference in 20 miles on a wet day and 200 miles on a dry one. Power steering doesn't conceal these differences but rather accentuates them. Indeed, some of the complaints levelled at power steering are actually due to lubrication.

We are now trying some experimental greases which appear to perform satisfactorily in dry weather for 800 to 1000 miles. From results thus far there seems to be no reason why lubricants could not be compounded to give consistent performance for at least 2500 miles.

So-called automatic greasing systems

add a multiplicity of parts susceptible to damage in order to introduce a lubricant that was inadequate in 1935. That doesn't seem to be the answer to the problem. The goal should be a superior lubricant introduced at less frequent intervals.

(Paper "A Critical Look at Chassis Lubricants" was presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 15, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Rheopectic Grease Answers Lube Problem

Based on paper by

J. D. NEESLEY
L. C. BRUNSTRUM
and
H. J. LIEHE
Standard Oil Company (Indiana)

NEW automatic chassis lubricators create lubricant problems. Oils feed well, but drip from the bearings; chassis greases feed badly, but they don't drip. What is needed is a lubricant that is both an oil and a grease. And a rheopectic grease—one that is fluid like an oil but acquires the consistency of a grease on passage through the lubricator—seems to fill the bill.

A rheopectic grease consisting of a lithium soap and a mineral oil of 400 SUS viscosity at 100 F will remain fluid in the lubricator reservoirs and in storage because agitation at low shear rate has no effect on consistency. The high shear rate obtained in the Multi-Luber system rapidly converts it to the consistency of an NLGI Grade 0 grease. (NLGI is National Lubricating Grease Institute.)

To check the stability and performance of this grease that flows like an oil, tests were conducted with an over-the-road tractor-trailer unit and a passenger car, both equipped with Multi-Lubers containing rheopectic grease. Observations were made for a period of five months, and throughout this time the grease in the lubricant reservoirs remained unchanged. Even at ambient temperatures as low as −20 F, flow was satisfactory and the lubricator functioned properly. When temperatures rose to 100 F, no evidence of drippage from the chassis under static conditions was observed.

(Paper "A Rheopectic Grease for Chassis Lubrication" was presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 15, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

All-Purpose Grease Still to Be Developed

Based on paper by

J. M. STOKELY

California Research Corp.

THE only true way to test grease is in the field. Some large users are attempting to write multipurpose grease specifications in terms of laboratory tests, but that only leads to elimination of some of the best performing greases.

Over 30,000,000 miles of passenger car service and 3,000,000 miles of heavy duty service coupled with laboratory tests show that most commercial multipurpose greases are adequate for light and medium duty service. One or two products compare favorably with specialized lubricants even in the heavy duty field. But no grease, multipurpose or specialized, is completely satisfactory under all field conditions.

If a multipurpose grease were developed which would approach the performance of specialized products in any particular service or fleet, it would be a great boon. It would save labor, equipment, and the time of dispensing. It would reduce inventories, and the product would be fresher. But until such a grease can be used in all vehicles under all conditions, it only adds one more item to an already large number of special products.

(Paper "Service Requirements for Multipurpose Automotive Greases" was presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 15, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Band Linings Offer Choice of Properties

Based on paper by

R. T. HALSTEAD

and

W. J. ECKERT

Johns-Manville Corp.

OF the several popular types of transmission band linings two predominate. One is the full molded, in relatively rigid segments with grooves; the other the flexible flat strip, supplied grooved or ungrooved. The former, once bonded to bands, is normally ready for installation; the latter requires the band-lining assembly to be bored to the drum diameter and the lining grooved before bonding.

Both types have distinct advantages and disadvantages. Full-molded segments, being rigid, are hard and resistant to physical deformation. They are used often when a thicker friction material is needed. The groove pattern which may be changed from segment to segment, and the gap between segments affords an escape route for the oil when the lining is applied to the drum. Segments are more difficult to coat with cement than flat strips and more difficult to position in the band during bonding. The adhesive tends to accumulate between segments during bonding and it is hard to remove.

The grooved, flexible, single-strip material is easier to process through the bonding operation. Its use reduces the chance of poor drum contact due to variations in lining thickness. Normally, the band-lining assembly is not bored because it would result in variable groove depth and permit no compensation for out-of-round bands. Grooved, single-strip linings must be

flexible enough to conform readily to the curvature of the metal band.

Ungrooved, single-strip linings lend themselves to boring operations to provide a precision band-lining assembly. The dimensional accuracy is unaffected by variations in thickness of the lining or of the cement film, or band out-of-roundness.

Good machining quality is desirable and can be built into the friction material to some degree. It influences selection of the fiber length. Some friction particles and other ingredients cannot be used because they chip, flake, or fall out during machining or finishing. If a lining machines poorly the finished surface will be rough and frequently pitted. The surface will wear more rapidly and will require more initial adjustment when the lining finally seats.

Good bonding qualities are important. Many band fabricators use heated anvil-type bonding machines which transmit heat through the friction material to the bond line. For this reason, linings must be able to withstand high bonding temperatures without blistering or deterioration.

(Paper "Certain Aspects of Friction Materials and Their Performance in Automatic Transmission Oil" was presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 16, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

Greases Need Built-In Water Resistance

Based on papers by

J. A. BELL

Shell Oil Co

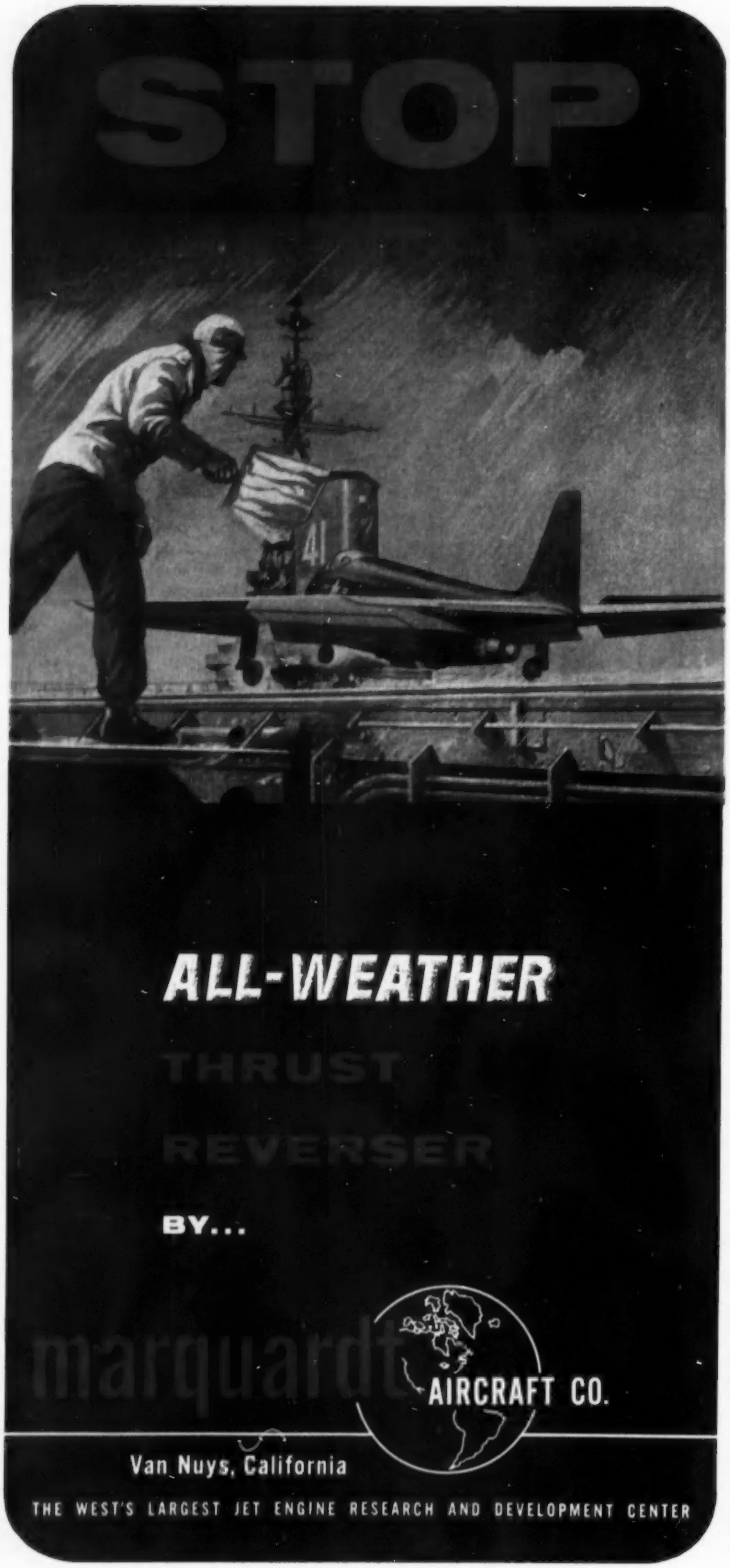
and

J. B. BELTZ

Oldsmobile Division, General Motors Corp

GREASES with built-in corrosion inhibiting properties will minimize the damage to chassis parts and wheel bearings resulting from water contamination. A 50-hour wet test of conventional and corrosion inhibited greases on an oscillating friction machine has shown the latter to give approximately twice the effective life of the conventional grease under wet operating conditions.

Evaluation of more than 100 conventional greases in an oscillating friction machine showed that all permitted some corrosion. Depending on the particular formulation, wet operation reduced the life of the grease by a factor ranging from 2 to 10. Only when greases were specifically formulated for corrosion resistance was com-



plete protection obtained for bearing surfaces.

Water does enter suspension joints and trouble does develop unless grease has water-resistant qualities. This was shown when cars were run repeatedly through a water bath, then their suspension joints disassembled and examined. From this experiment it was concluded that better performance life of chassis greases can be obtained by an increase in the viscosity of the base oil from which the grease is compounded. The resultant increase in front-suspension friction, however, must be kept within reasonable bounds. There is now need for a grease having a flat viscosity curve, with good performance life at elevated temperatures, and without excessive friction at low temperature.

(Papers "Corrosion Inhibited Automotive Greases" (Bell) and "Chassis Grease Retention Tests" (Beltz) were presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 15, 1955. They are available in full in multilith form from SAE Special Publications Department. Price: 35¢ each paper to members, 60¢ to nonmembers.)

Centralized Chassis Lubrication Advances

Based on paper by

CARL H. MUELLER

Lincoln Engineering Co.

CENTRALIZED lubricating systems fall into two general categories: those which measure the lubricant at the source and those which measure it at the bearing. Then there are variations of the basic types, such as high pressure, low pressure, positive displacement, feed regulation by restriction, and others.

Tests of centralized systems on several fleets lead to the conclusion that a system for trailer lubrication should have the following characteristics:

1. Measurement at the source in order to provide an individual line from the pump to each bearing. If a line breaks then only the associated bearing suffers.

2. Positive displacement to insure equal delivery to all bearings regardless of resistance offered by individual bearings.

3. High pressure to insure lubricant delivery to bearings at low temperatures when line resistance is high.

4. Fully automatic operation requiring no driver attention. An inter-

lined trailer may be pulled by half a dozen drivers before it returns to home base, making difficult the passing on of instructions.

Centralized systems have been tried out in a number of passenger cars and have operated satisfactorily. However, installation costs have been regarded as excessive. Today, the situation is changed, due to two developments —ball joint suspensions and a new type of nylon tubing. The ball joint suspension has fewer bearings, making it possible to service a car with one pumping unit, while the tubing provides low cost, durable, flexible connections. This nylon tubing makes possible fabrication of an easily installed tubing harness similar to the familiar electrical harness.

(Paper "Centralized Chassis Lubrication" was presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 15, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

1000-Mile Greasing Proved Need by Test

Based on paper by

J. F. McGROGAN

The Atlantic Refining Co., Inc.

COMPLAINTS of chassis squeaks are numerous in the first 250 miles of driving following a grease job, but they really soar after a car has gone 1250 miles. And it doesn't make much difference what background the drivers have. All recognize squeaks for what they are.

These are the findings of a road test, using six cars each of two different makes, to investigate the influence of grease composition on chassis performance.

Company employees drove the cars day after day as they would be driven in owners' hands. Nevertheless, despite the technical background of the drivers, the number of complaints of squeaking differed little from that registered by other groups, being 4.8 per test driver and 4.5 for the general public.

Although the test indicates the importance of the 1000-mile greasing period, it also shows the need to improve chassis greases.

(Paper "Chassis Lubricant Performance—Driver's Reaction" was presented at SAE Golden Anniversary Summer Meeting, Atlantic City, N. J., June 15, 1955. It is available in full in multilith form from SAE Special Publications Department. Price: 35¢ to members, 60¢ to nonmembers.)

About SAE Members

continued from page 91

Warner's research, manufacturing, and sales activities on the Pacific Coast.

SEYMOUR R. KROWN is now video recording device engineer with National Broadcasting Co., Inc., Hollywood. He was television technical director, U. S. Army, Signal Corps Pictorial Center, Long Island City.

DONNELL R. SULLIVAN is now associated with Dearborn Marine Engines, Inc. as sales engineer. He has been serving as junior design engineer for Ternstedt Division, General Motors Corp.

WILBUR E. ROBBINS has taken the position of chief chemist with Caribbean Refining Co., Bayamon, Puerto Rico. He had been Head, Fuels Division, Fuels & Lubricants Project, Department of the Navy, Naval Engineering Experiment Station, Annapolis, Md.

HOWARD VOGEL has recently been appointed manager of the Blue Crown Spark Plug Division, Blue Crown Spark Plug Corp. He had been chief engineer for all plants of the Champion Spark Plug Corp.

Bartlett

W. H. BARTLETT is now vice-president, Manufacturing, for Dunlop Canada, Ltd. He has been serving as general works manager.

Beckett

RONALD C. BECKETT has been appointed chief engineer of the Oil Filter Division, Michiana Products Corp. He has specialized in oil filtration since 1944 and came to Michiana from the Oil Filter Division of Walker Mfg. Co., where he was chief engineer.

WILLIAM W. HURTT is now associated with Boeing Airplane Co., Wichita Division, as technical assistant to the chief of development. He had been assistant chief engineer—Design with Temco Aircraft Corp., Dallas, Texas.

GERALD REINSMITH is now acting Chief of the Research Branch, Research and Development Division, Ord-

About SAE Members

continued

nance Weapons Command Headquarters, Rock Island, Ill. He had been factory manager of Narmco Metlbond Co., Chino, Calif.

JOHN W. THOMPSON, who has been aircraft and maintenance engineer for Parsons, Brinckerhoff, Hall & MacDonald, New York, has joined Curtiss-Wright Corp., Caldwell Wright Division, as sales manager.

D. G. PROUDFOOT has been elected vice-president of Pennzoil Co. He has been serving as manager of lubrication sales.

Pennzoil has recently merged with South Penn Oil Co.

Hitchcock

CLARE L. HITCHCOCK is now sales engineer of Studebaker truck department. He will serve as sales liaison with engineering and as administrative assistant to the Studebaker Division truck department general manager.

GEORGE J. PAYNE, previously specification engineer for Detroit Edison Co., is now test and development engineer with Chrysler Corp. Engineering Division Missile Branch.

FOSTER R. GAYLORD has joined Ryder Truck Rental System, Inc., Atlanta, Ga., as branch manager. He had served in the same capacity with Reo Motors, Inc., St. Louis, Mo.

S. A. STRICKLAND has been named vice-chairman of the board of the new Federal-Mogul-Bower Bearings, Inc., Detroit.

DR. ARTHUR A. BROWN is now associated with Pratt & Whitney Aircraft Division, United Aircraft Corp., as project engineer. He had been vice-president for Frederick Research Corp., Bethesda, Md.

LAWRENCE A. BEILBY, previously district manager, General Motors of Canada, Ltd., is now Royal Canadian electrical and mechanical engineer with the Department of Defense, Halifax, Nova Scotia, Canada.

ROBERT D. SIDEL is now associated with Ryder Truck Rental System, New York City, as executive vice-president. He has been serving in the same position with Metropolitan Distributors, Inc. of New York.

SAMUEL RAY, formerly eastern sales representative for Chicago Forging & Manufacturing Co., has taken a position with Trylon Machine and Gear Co. of North Bergen, N. J., as sales manager.

New Members Qualified

These applicants qualified for admission to the Society between July 10, 1955 and August 10, 1955. Grades of membership are: (M) Member; (A) Associate; (J) Junior.

Atlanta Section

Arthur A. Kizer (A).

Canadian Section

Charles B. Benton (A), P. A. Courage (A), James D. Dixon (A), John B. German (A), Theodore M. Green (M), A. C. Jamieson (J), Ralph K. Shantz (A), John M. Wachsmuth (A).

Central Illinois Section

Paul C. Rosenberger (J), Harold C. Wiggers (J).

Chicago Section

Walter Ambs Forrest (A), Harold E. Greenlee (M), K. E. Grundvig (M), Jeremiah Halpin (M), James H. Kahlke (M), Harold E. Pickles (A), Robert F. Potteiger (A), H. T. Thompson (A), Earl C. Uban (J), George R. Willy (A), Herman M. Winkler (A).

Cincinnati Section

Stanley Braun (J), H. A. Fremont (M), Willard G. Snyder (A).

Cleveland Section

Clarence Harold Culler (M), Earl G. Cummings (M), E. W. Husemann (M), Whitmell T. Rison (M).

Dayton Section

William C. Posey (J), W. R. Smith (M).

Detroit Section

Joseph F. Balgenorth (A), Thomas H. Bartlett (A), Louis G. Bettega (A), V. E. Boehne (M), Frederick J. Button, Jr. (J), Richard B. Chapin (A), Stanley D. Cockburn (J), Thayer M. Cowman (A), Carl J. Demrick (M), Dan D. Epps (A), J. T. Galvin (A), Bernard G. Golden (A), Daniel William Hall (J), James H. Henry (J), Donald A. Jones (J), Joseph Kwaselow (M), Bernard M. Lacny (J), Robert J. Lauer (M), Jack O. Lefton (A), Walter J. Miller (M), Jack E. Morgan (J), Lester V. Ostrander (M), Fred E. Parsons (A), George J. Porth (J), Paul A. Ribaux (M), Harold H. Schroeder (M), Arthur J. Smolinski (J), Rene H. Vansteenkiste (M), William W. Vincent (M), William Weightman (M), Calvin G. Wells (J).

New Members Qualified

continued

Indiana Section

Basil G. Bentz (M), Percy L. Ford (A), Philip L. Fosburg (M), Oliver T. Nephew (J).

Kansas City Section

Robert L. Wells (M).

Metropolitan Section

Michael Francis DeMarco (J), Rudolf L. Graupe (M), Robert Nelson Hagarty (M), Jack H. Kasley (M), Jack J. Lichtenberg (M), Robert E. Loren (M), Lt. John Holmes McCalla (M), Milo F. McCammon (M), Walter D. McKinley (M), Glenn E. Newman (M).

Mid-Continent Section

Bert Samuel Davenport (M).

Mid-Michigan Section

Harry J. Erickson (M), Frederick W. Hyslop (M), Max Phillips (M), F. Eugene Smythe (M), Jack R. Wallace (M), William D. Wolfe (M).

Milwaukee Section

R. L. Burks (J), Edgar L. McFerren (M), Gale E. Nordstrom (M), Harold F. Pokorney (M), Steve A. Ropar (J).

Montreal Section

Lennox L. Joseph (A), Alain Prefontaine (A).

New England Section

Frank Cragg (M), George R. Winkler (J).

Northern California Section

Lloyd L. Davis (A), C. P. Johnson (A), Arthur L. Lyman (M), B. F. Poduska (A).

Northwest Section

Harold R. Ginther (J), W. W. McFadzen (A).

Philadelphia Section

Thomas P. Demuth (M), Berton Karol (M), George Murray Powell, Jr. (M), Charles B. Rawson (M), William Schreiber (M).

St. Louis Section

Alexander P. Fox (M), Robert W. Keller (A), Gerald C. Maechler (M).

San Diego Section

Ray Bert Kalanquin (A).

Southern California Section

Francis B. Murphy (A), William A. Rothwell (J), Sumner B. Sargent (M), Horace M. Skinner (J), William L. Stabler (A).

Southern New England Section

John Burwell Beckwith (M), Martin J. Glenday (J), Glen A. Guernsey (M), Edmond H. Judd (J).

Spokane-Intermountain Section

Russell J. Pohl (A).

Texas Section

Martin G. Flood (J), David H. Norton (M).

Texas Gulf Coast Section

John A. Scofield (M).

Virginia Section

Elwood C. Orrell, Jr. (A).

Western Michigan Section

Edsell Martindale Eady (M), Harvey A. Matthews (A), Clifford J. Walsh (M).

Wichita Section

John J. O'Neill (M).

Williamsport Group

David I. Jenkins (M), Alfred R. Klinger, Sr. (A).

Outside Section Territory

Kenneth R. Harman (M), Sven E. Johannson (J), John C. Rundell (A), Edmund V. Wesely (A).

Foreign

Max John Burkhard (M), Ceylon; H. Jose J. d'Ornellas (M), Peru; Kshitindra Kanta Ganguli (J), India; J. L. E. Groff (M), France; Marcos Hadjes (J), Brazil; Walter Geoffrey Hall (M), Australia; Clarence Hoogerwerf (M), Holland; Sami M. Kazes (J), France; Feroze K. Viccaji (J), India.

Photo by Sarra Inc.

Applications Received

The applications for membership received between July 10, 1955 and August 10, 1955 are listed below.

Baltimore Section

Donald H. Groft.

British Columbia Section

George T. Perry.

Buffalo Section

Robert D. Snyder.

Canadian Section

M. B. Jackson, Theodore J. Kish, Peter B. Mackenzie, Duncan M. McLaurin.

Central Illinois Section

Charles L. Ellis, Randolph E. Seyfried.

Chicago Section

Richard J. Lambert, Carl A. Loy, Jack Mitchell, Richard P. Molloy, Elliot S. Nachtman, Elden E. Schott, Paul B. Stephens, Otmar E. Teichmann.

Cincinnati Section

Clarence R. Wuellner.

Cleveland Section

Joseph August, Jr., James B. Hoffman, James G. Burrows, J. J. Pirnat, R. E. Saxton, James H. Simler.

Colorado Group

Lee R. Sollenbarger.

Dayton Section

Arthur Frank Pelster.

Detroit Section

Hulki Aldikacti, Robert T. Bergman, Robert B. Boswell, R. Thomas Bundorf, Richard L. Courtney, Frederick R. Dennison, Frank Dickenbrock, Jr., Howard G. Foell, Ivan M. Francis, Hilmar W. Haenisch, Donald R. Hannum, Ernest C. Harris, William J. Herrmann, Richard E. Hinze, Gibson O. Hufstader, Conrad Kaspers, Carl Edmund Kime, Murdoch J. MacKenzie, Frank E. Manion, John J. Mears, Ralph H. Merkle, James J. Murtagh, William O. Murtagh, Henry A. Nickol, James H. Orrick, G. B. D. Peterson, Michael E. Quinn, Emerald G. Schenk, Lowell C. Schneider, Louis W. Schultz, Robert S. Siegler, Paul S. Silber, Preston T. Tucker, Jr., John S. Wagoner, Doyle Warner, Jr., Julian J. Woytowich.

Hawaii Section

Thomas R. H. Lillie, Clifford D. Yancey.

Indiana Section

Alfred G. Beier, Robert Highland, Jack Ryan, Jr.

Metropolitan Section

Henry E. Buttelmann, Bruce P. Fow, Howard B. Huntress, A. Vincent Maisoneuve, Sol Mendelson, James P. Moreland, J. Thomas Morrow, Henry Voorman, Jr.

Mid-Continent Section

Daniel M. Gibson, Jr., Fritz W. Weilmuenster, Jean R. Whitlock.

Mid-Michigan Section

Robert L. Kersten, Harold W. Noponen, Gerald J. Schmidt.

Milwaukee Section

Royce N. Brown, K. Mahadevan, William A. Riebe, Russell K. Sesto.

Montreal Section

Ian C. Colquhoun, Stanley C. McRobert, James Rankin.

New England Section

John D. Morrison.

Northern California Section

Frederick S. Bonney, Robert D. Firth, Robert LeRoy Pohl, Robert E. Totman.

Northwest Section

Donald H. Niesse, Emerson F. Reiber, William S. Stirling.

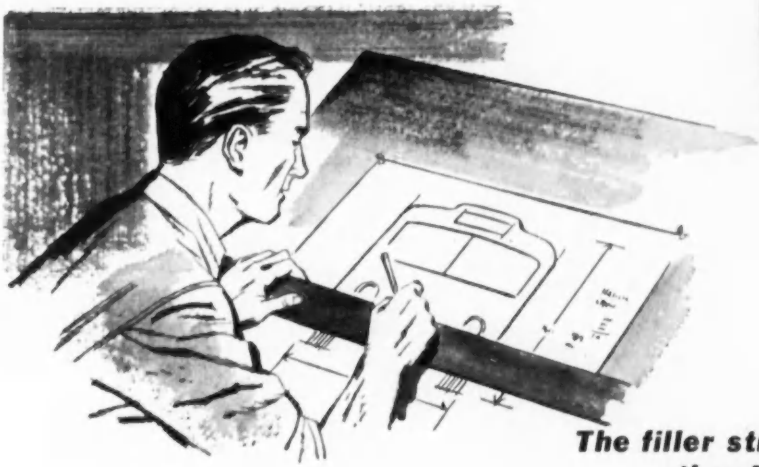
Oregon Section

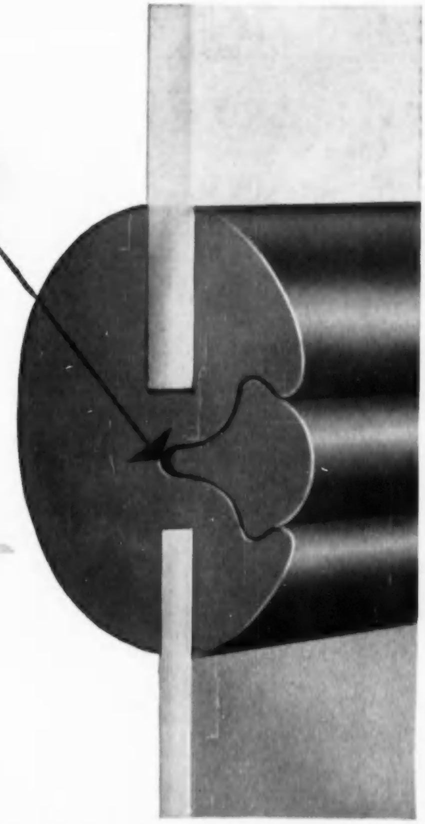
Dewey H. Campbell, Harry W. Dressler.

It's this separate

FILLER STRIP

that puts more pressure on the fence and the glass . . . allowing new freedom of design!





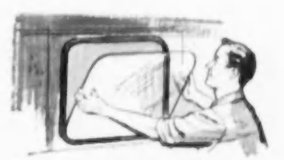
The filler strip makes possible these other Inland advantages!

Today, weather stripping is a "snap". And, it's the snap of Inland's new weather strip that does it! Just think of the design freedom you have—now that weather strip problems are gone. With Inland Self-Sealing Weather Strip, *you need make no provisions for cement, clamps, moldings or channels. You are assured a water-tight seal every time!*

The patented Inland Filler Strip enables the installer to compress the sealing strip *after* the glass is in place. And because it eliminates all the head aches of trying to force the glass into a compressed groove, this is the easiest weather strip to install!

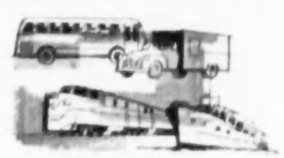
INLAND MANUFACTURING DIVISION
General Motors Corporation • Dayton, Ohio

LEAK PROOF! Permanently leak proof, because it seals both glass and body panel under powerful compression.



EASY GLASS REPLACEMENT! Less lost time for vehicles—broken glass can even be replaced on the road, if necessary.

A POSITIVE SEAL! New filler strip puts more pressure on the fence and the glass—assures complete positive weather proofing every time.



VERSATILITY! Ideal for vehicles, booths, trains, gasoline pumps, buildings, marine windows—for positive, permanent sealing of any window or panel!

INLAND MANUFACTURING

Self-Sealing WEATHER STRIP

Applications Received
continued

Philadelphia Section

Robert N. Brower, Robert P. Burns, Victor D. Hajj, Alfred D. Kochel, Paul E. Oberdorfer, Jr., Richard R. Pruyn, Stanley S. Wulc.

Pittsburgh Section

Howard M. Bernbaum, Charles G. Purnell.

St. Louis Section

Norman B. Hefti, Earl F. Hubacker, Jr., Frank E. Pipe, George M. Stonum.

San Diego Section

Leonard J. Fiorito, Ted A. Matera.

Southern California Section

Charles D. Babb, John Carrigan, Fred A. Cohan, James E. Fink, Harold Greer, Roger E. Lagerquist, Harlie LeRoy Lunke, Fred Murg, Robert Leo O'Loughlin, G. Gordon Prentice, J. Bruce Ramsey, Herman V. Schwalenberg, Robert Simmons, Frank Sperl, Jr., James C. Spracher.

Southern New England Section

Robert W. Martin, Oliver E. Spencer.

Syracuse Section

Warren M. Meaker.

Texas Section

Charles K. Cates, Louis L. Finch, Harvey E. Fisk, III, A. F. Love, Jr., Preston D. Megginson, Glen D. Snyder, Edwin F. Thomas.

Texas Gulf Coast Section

W. H. Heesche, Travis R. Shipp.

Twin City Section

Carlos A. Claure, Paul S. Petersen, Douglas M. Skoglund.

Virginia Section

Charles J. Gaetz.

Western Michigan Section

Curtis E. Behrens, Jack R. Osborn.

Wichita Section

Albin H. Swanson, Jr., William A. Swope.

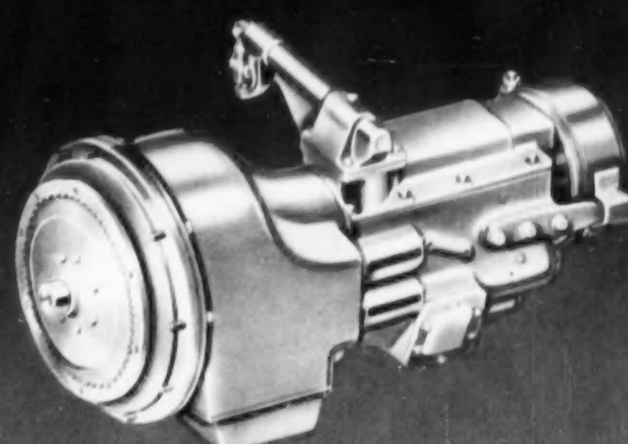
Outside of Section Territory

Anthony Deltufo, James K. Draper, Henry L. Marple, E. F. Meier, Jamil A. Razzak, Philip G. Rector, Glen E. Smelcer, Jack Weinroth.

Foreign

James Alcock, England; John Hamilton Lee, Australia.

Vastly reduced shifting with TORCON Package Torque Converter-Transmission

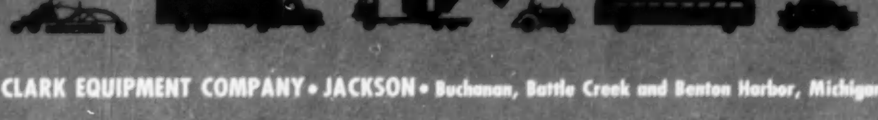


You profit substantially by using this TORCON "package" torque converter and transmission:

- operational shifts vastly reduced
- 3-to-1 torque multiplication as needed—smooth, cushioned power
- longer life of transmission
- maintenance greatly reduced by fewer shifts, longer life

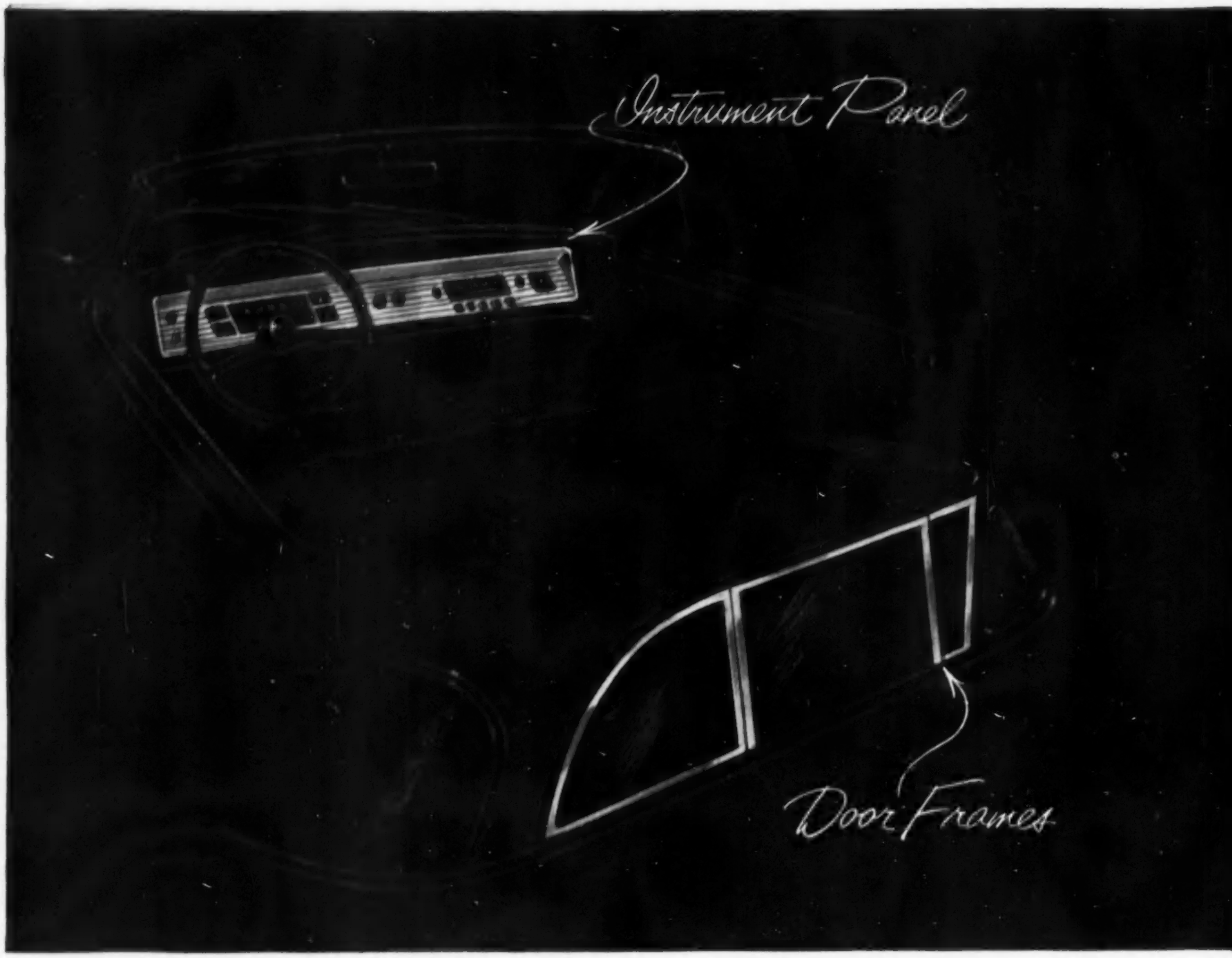
TORCON'S broad line meets every power-transmission need.

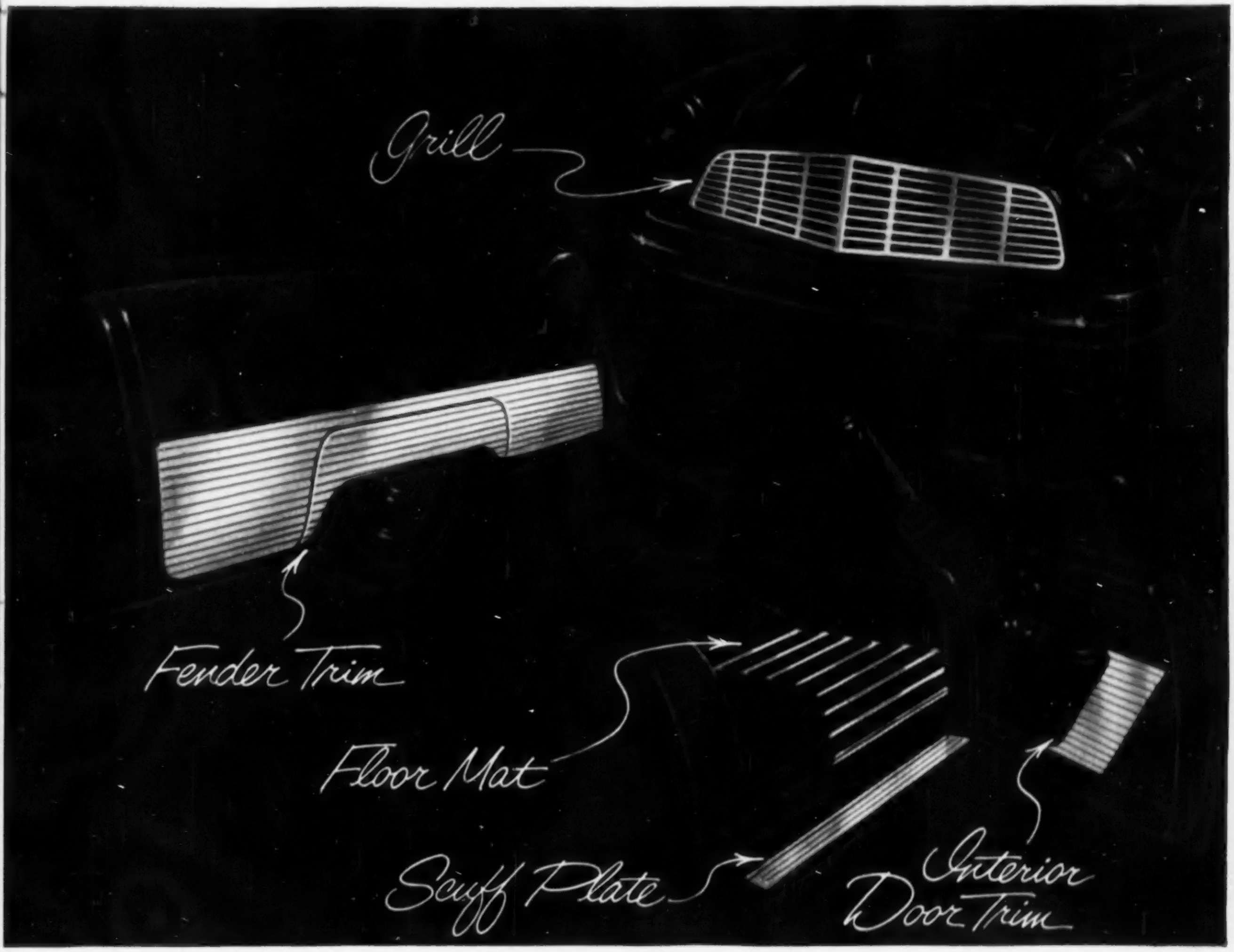
You'll find the Torcon bulletin helpful.



CLARK EQUIPMENT COMPANY • JACKSON • Buchanan, Battle Creek and Benton Harbor, Michigan

CLARK® EQUIPMENT


Instrument Panel
Door Frames

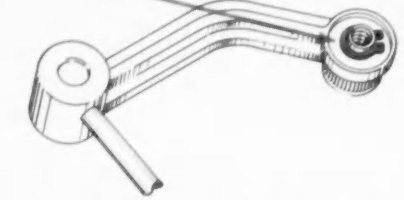

Grill
Fender Trim
Floor Mat
Scuff Plate
Interior Door Trim

Waldes Truarc Rings Permit Better and More Economical Design—Fewer Parts, Faster Assembly, Minimal Rejects!

Viewlex's Change-O-Matic

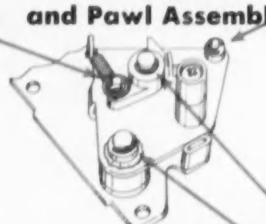
Viewlex, Inc., L.I.C., N.Y. solves a variety of fastening problems in their new model Change-O-Matic automatic slide changer. Assembly time is speeded, parts eliminated, rejects lowered, and compact, economical product design achieved.

Connecting Arm Assembly



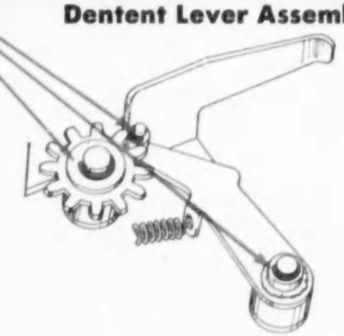
Use of one Waldes Truarc Ring (Series 5100) eliminates time-consuming riveting over the flange, retains and holds the connecting knob captive. A compact, neat design is made possible, rejects caused by poor riveting eliminated.

Actuating Plate and Pawl Assembly



Three types of Waldes Truarc Retaining Rings are used in this assembly. One circular self-locking ring (Series 5105) retains pawl return spring. One external E-ring (Series 5133) retains roller used to prevent gear motion during transport cycle. One crescent ring (Series 5103) retains pawl which indexes gear. Second crescent ring retains sub-assembly. Truarc Rings speed assembly, cut rejects, eliminate parts.

Dentent Lever Assembly



Truarc E-ring (Series 5133) eliminates use of tapped hole and shoulder screw, retains roller which prevents over-travel of gear. Assembly is rapid.

Whatever you make, there's a Waldes Truarc Retaining Ring designed to improve your product . . . to save you material, machining and labor costs. They're quick and easy to assemble and disassemble, and they do a better job of holding parts together. Truarc rings are precision engineered and precision made, quality controlled from raw material to finished ring.

36 functionally different types . . . as many as 97 different sizes within a type . . . 5 metal specifications and 14 different finishes. Truarc rings are available from 90 stocking points throughout the U. S. A. and Canada.

More than 30 engineering-minded factory representatives and 700 field men are available to you on call. Send us your blueprints today . . . let our Truarc engineers help you solve design, assembly and production problems . . . without obligation.

For precision internal grooving and undercutting . . . Waldes Truarc Grooving Tool!

Send for new catalog supplement

WALDES
TRUARC®
RETAINING RINGS

Waldes Kohinoor, Inc., 47-16 Austel Place, L. I. C. 1, N. Y.
Please send the new supplement No. 1 which brings Truarc Catalog RR 9-52 up to date.
(Please print)

Name ______
Title ______
Company ______
Business Address ______
City ______ Zone ___ State ______

SA-097

WALDES TRUARC Retaining Rings, Grooving Tools, Pliers, Applicators and Dispensers are protected by one or more of the following U. S. Patents: 2,382,948; 2,411,426; 2,411,761; 2,416,852; 2,420,921; 2,428,341; 2,439,785; 2,441,846; 2,455,165; 2,483,379; 2,483,380; 2,483,383; 2,487,802; 2,487,803; 2,491,306; 2,491,310; 2,509,081; 2,544,631; 2,546,616; 2,547,263; 2,558,704; 2,574,034; 2,577,319; 2,595,787, and other U. S. Patents pending. Equal patent protection established in foreign countries.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Remember
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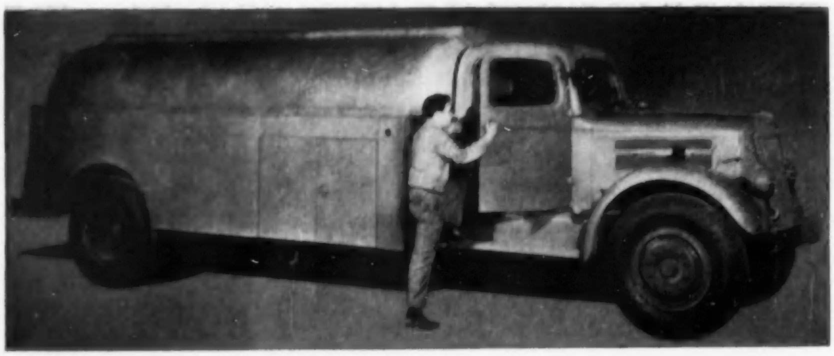
R/M
FIRST IN FRICTION

HOW R/M ENGINEERING SETS

These are just a few of the friction parts—brake linings, clutch facings and automatic transmission bands—that R/M makes of woven asbestos.

THE RECORD OF "FIRSTS" IN FRICTION MATERIAL DEVELOPMENT SHOWS WHY R/M IS FIRST IN FRICTION

FIRST Woven Brake Lining • FIRST Asbestos Brake Lining • FIRST Ground Wearing Surface • FIRST Zinc Alloy Wire Brake Lining • FIRST Pre-Treated Yarns • FIRST Extruded Pulp Brake Lining • FIRST Flexible Pulp Brake Lining in Rolls • FIRST Dry Process Brake Lining • FIRST Semi-Metallic Brake Lining • FIRST Bonded-to-Metal Brake Lining • FIRST Woven Clutch Facings • FIRST Molded Asbestos Clutch Facings for Clutches Operating in Oil FIRST Endless Woven Clutch Facings • FIRST Pre-Treated Clutch Facings • FIRST Bonded-to-Metal Clutch Facings



Deadweight was reduced 600 lbs. in the shell of this metropolitan delivery truck-tank unit through the use of Mayari R. The tank, which was built for an oil company in California by the Independent Iron Works, Oakland, was made of 12-gage Mayari R sheets and weighs 3200 lbs. The capacity of the tank is 2200 gal.

This fleet of 2-ton trucks holds down both operating and maintenance expenses, thanks to reduced deadweight in bodies built of Mayari R by Leonhardt Body Division of Charles T. Brandt, Inc., Baltimore, Md. Furthermore, life of truck bodies is greatly extended by the corrosion-resistance of Mayari R, a high strength, low alloy nickel steel, produced by Bethlehem Steel Co., Bethlehem, Pa.

Nickel alloy lengthens truck body life ...allows 15% weight reduction

These truck bodies are made from Mayari R...a high strength, low alloy steel containing nickel.

Its use in these bodies enabled the builder to cut deadweight approximately 15 per cent. Because thin, light sections of this type steel provide the same strength as thicker, heavier sections of plain carbon steel.

What's more, Mayari R and similar high strength, low alloy nickel steels offer 5 to 6 times the corrosion resistance of carbon steel in atmosphere. And 2 to 4 times that of copper bearing steel.

As a result, these steels retain most of their original section thickness and hence their original strength during years of use. This means less maintenance . . . and greatly lengthened equipment life. Their resistance to impact and abrasion is also superior to that of carbon steels.

Get the facts...send for a copy of "Nickel-Copper High-Strength Low-Alloy Steels." Cover-to-cover, it's full of valuable data. It tells you how these nickel steels perform on a variety of jobs...their response to fabrication . . . and design factors that can help you cut deadweight. Write for your copy now and learn how these steels can save you money.

INCO **THE INTERNATIONAL NICKEL COMPANY, INC.** 67 Wall Street New York 5, N.Y.

moraine engineering

A new and improved spacer for shock absorbers that is now being made in one press operation— with substantial economies for a Moraine customer.

Amazes even engineers!

People sometimes find it hard to believe us when we tell them that parts like these can be made from metal powder in one press operation. Then they're amazed to see us make these and other equally complicated parts easily, economically, in quantity. And when they learn about the over-all advantages — *Improved product performance . . . Lower unit cost . . . Dependable, on-time quantity delivery . . . Engineering assistance . . . Savings in customer investment in equipment and floor space* — they are convinced that Moraine engineering and metal powder techniques may offer real solutions to their own problems.

Moraine products include: Moraine-400 bearings, toughest automotive engine bearings ever made— Moraine friction materials — Moraine metal powder parts—M-100 engine bearings and Moraine conventional engine bearings—Self-lubricating bearings—Moraine porous metal parts —Moraine power brakes—Delco hydraulic brake fluids—Delco master cylinders, wheel cylinders, lined brake shoes and parts.

moraine products

DIVISION OF GENERAL MOTORS, DAYTON, OHIO

How Great Lakes Steel *looks* at quality

SAMPLE "PINS" from heats are sent to a special Quality Control Laboratory where analyses of previous tests by wet chemistry are double checked by spectrograph.

SPECTROGRAPH is used to make doubly certain the finished steel meets the customer's specifications. Here a densitometer reading is made of a spectrogram to determine the percentage of elements present in the steel.

Quality is something you can *see* in our modern laboratories. In the photograph above a spectrogram is readied for reading in the densitometer—and one more test is underway to help assure quality.

Precision control tests such as this one are applied at every stage of production to assure you the quality of steel required for your product and production methods.

At Great Lakes Steel, the emphasis is on quality and service. Where your production problems involve steel—and particularly flat-rolled steel—we invite you to make them *our* problems. Great Lakes Steel is as close as your telephone.

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ished steel meets the customer's specifications. Here a densitometer reading is made of a spectrogram to determine the percentage of elements present in the steel.

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IS OUR BEST FEATURE!

ROHR is the world's largest producer of ready-to-install power packages for airplanes . . . such as the Lockheed Constellation, Douglas DC-7, the all-jet Boeing B-52 and other great military and commercial planes. Result? A world of production experience and engineering skill!

Today, besides power packages, this skill and experience go into the making of over 30,000 other parts for aircraft of all kinds. For the aircraft parts you need . . . call on ROHR. Remember, thousands upon thousands of famous ROHR power packages, plus millions of other aircraft parts, have made experience our best feature.

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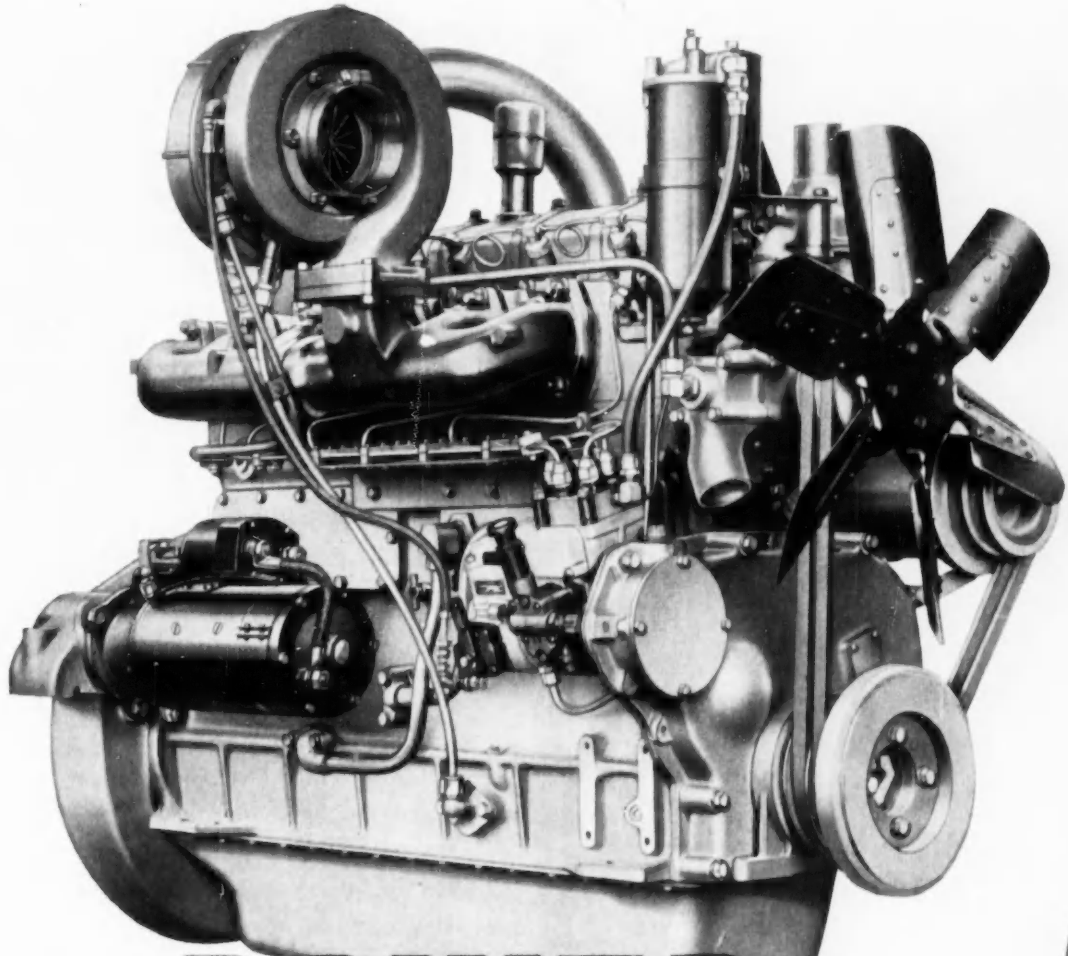
ROHR
AIRCRAFT CORPORATION

CHULA VISTA AND RIVERSIDE, CALIFORNIA



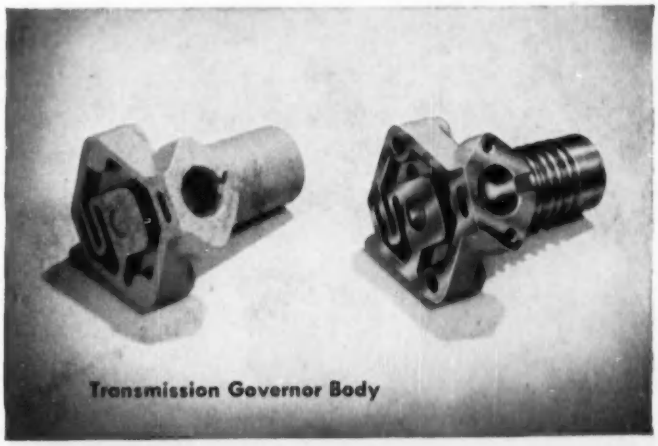
288

chine cycle time. On the combination camshaft and gear shown at right, preliminary rough cuts are not needed and the shell casting goes straight to the final machining operations. The uniformity of Central Foundry Division shell castings permits grinding without the preparatory roughing operations, thus reducing machine time and tool costs in this and many other applications.

Improved Casting Surfaces

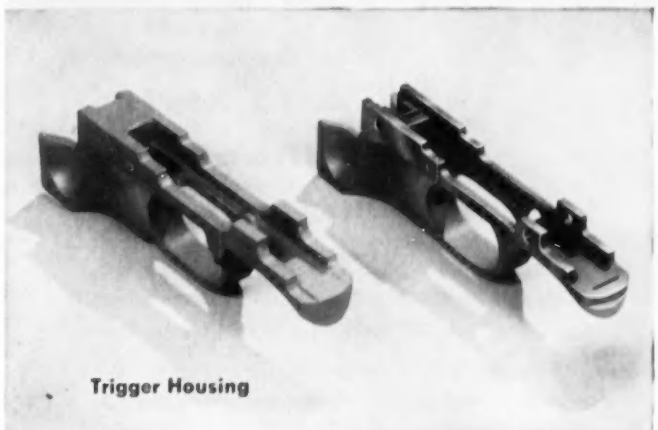
The improved casting surfaces and close tolerances of a shell casting often eliminate machining on non functional areas. Such is the case with the trigger housing of an automatic rifle, shown at lower right. The part was first produced from a forging but is now shell cast of Armasteel. The profile milling necessary on the forged part has been entirely eliminated and only critical dimensions are now machined.

For further information about the shell casting process and how it may improve or effect economies in your product, write for descriptive literature . . . or request personal help from our experienced engineers, without obligation.



Transmission Governor Body

Combination Camshaft and Sprocket

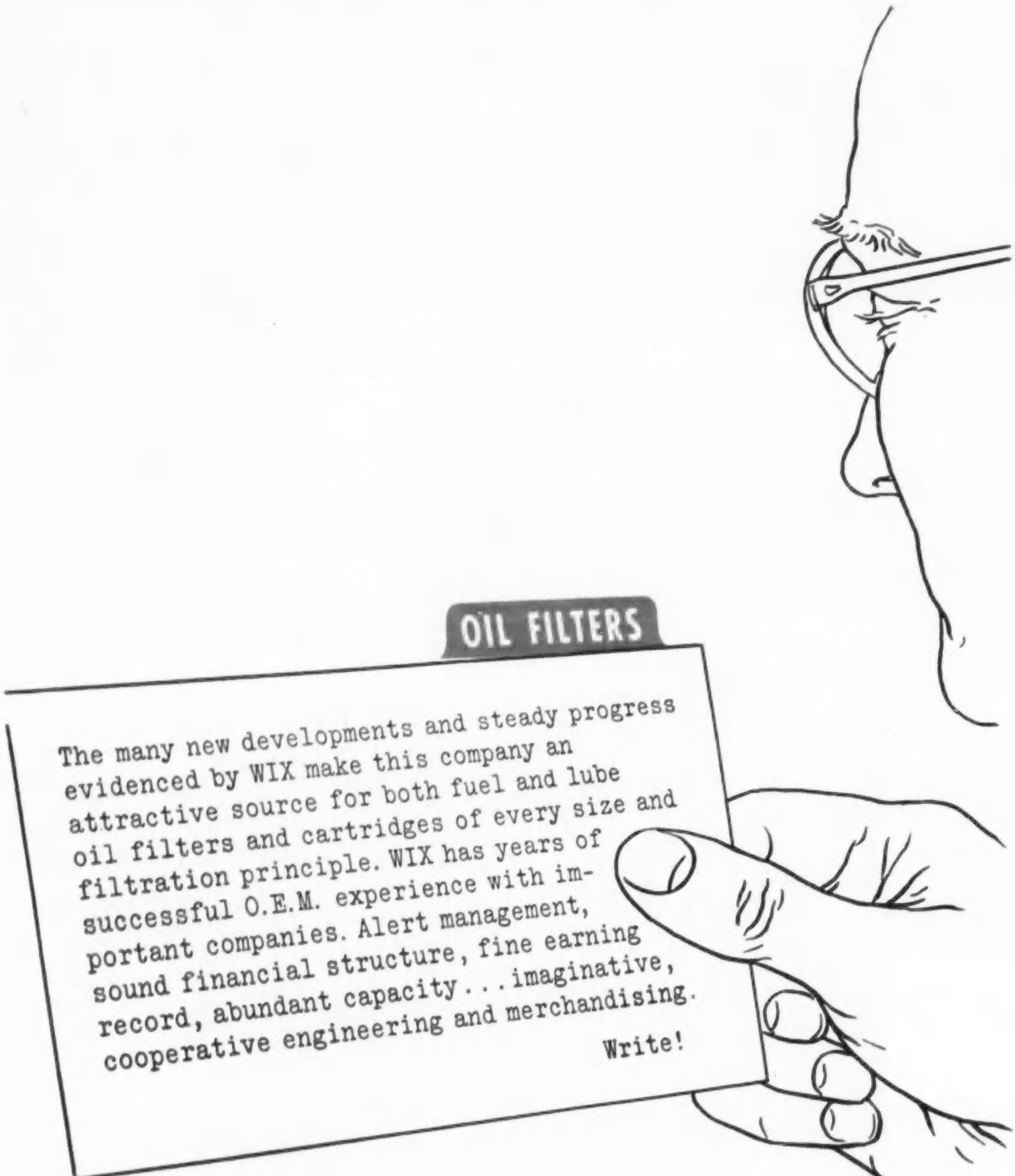


Trigger Housing

52

CENTRAL FOUNDRY DIVISION

GENERAL MOTORS CORPORATION • SAGINAW, MICHIGAN • DEPT. 18



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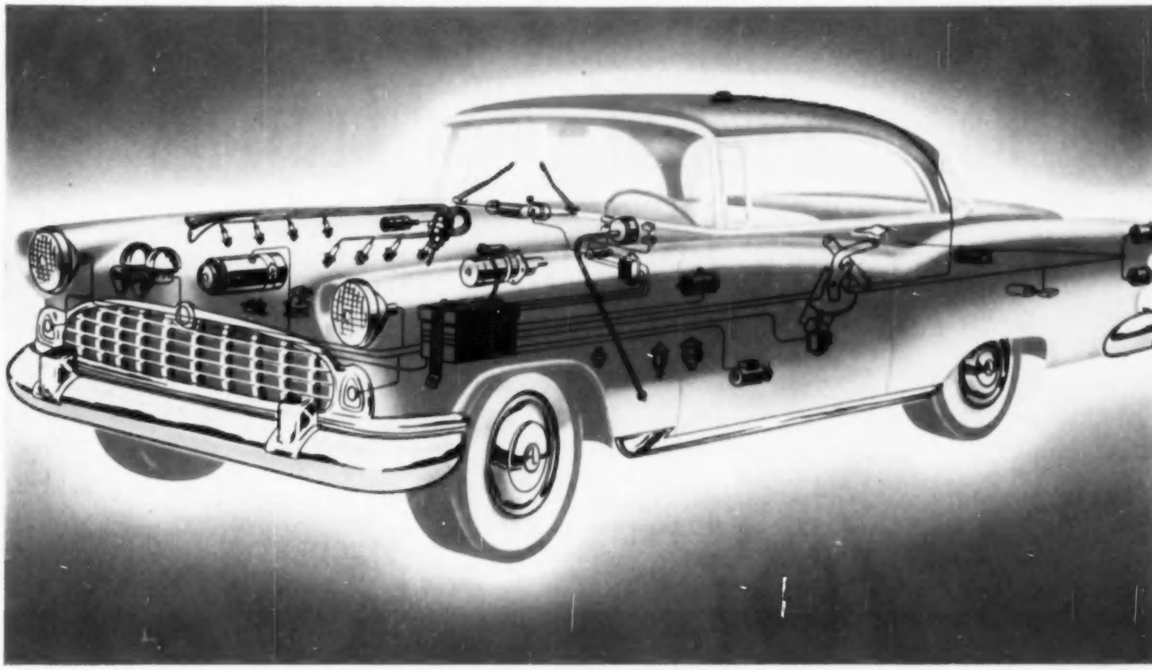

Let IPC take a look at your drafting board through a seal designed to meet your requirements.

Planning to manufacture new machinery? You can eliminate mechanical sealing headaches by asking IPC to design your oil seals and packings.

Dependable, trouble-free oil seals and packings are a direct result of good planning . . . and IPC design engineers have analyzed and developed cup, flange, U, V packings and oil seals for all kinds of industrial applications. Special synthetic or leather models, bonded case oil seals, O-rings, etc. will be drafted for *your specific problem*.

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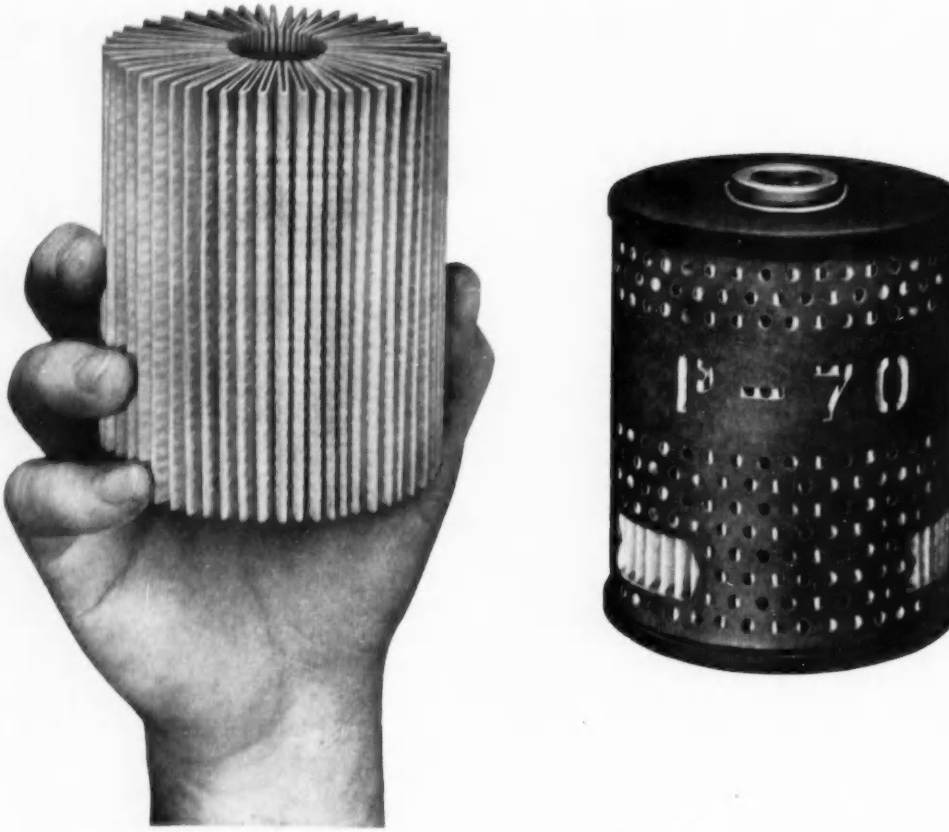

Here's the secret of Micronic*-type Purolator's

HIGH FLOW RATE

This little Purolator filter element can clean a quart of dirty lube oil in 60 seconds. It takes out sludge and solid impurities as small as one micron (.000039-inch) yet leaves beneficial additives unaffected. It operates with minimum pressure drop and a standard-size oil pump.

You can see the secret at the left. It's the Purolator Micronic element. This accordion-pleated, resin-impregnated element provides ten times the filtering area of older elements. This means faster filtration rates and far greater dirt storage capacity.

To designers and users of automotive equipment, Micronic-type Purolators offer thorough filtration by a small, compact unit that fits snugly into the lubricating system without needing an oversized pump to boost pressures through the filter. These advantages of Micronic filtration are among the many reasons why original automotive equipment manufacturers use more Purolators than any other type of filter.

Micronic elements do not channel. They are waterproof and warp-proof and remain unaffected by engine temperatures. There's a Purolator to fit every vehicle, tractor, and other gasoline- or diesel-engine-powered unit in service today. Write for our automotive catalog, No. 2054, to Purolator Products, Inc., Rahway, N. J., Dept. A3-917.

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DETROIT 12, MICH.—WINDSOR, ONT.

on Truck Tubeless Tire Valves

An Industry Development Keeping Abreast of Transportation Progress

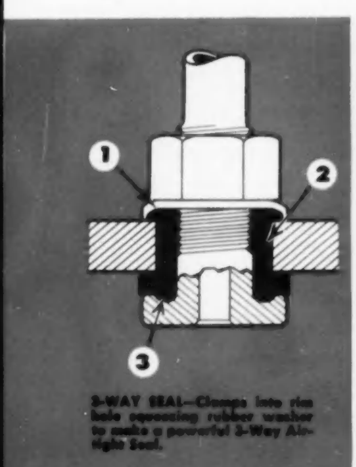
Truck tubeless tires are here. Recent approval of certain rim designs by the Tire and Rim Industry gives truck tubeless the green light. Commercial vehicle manufacturers indicate that tubeless tires will be mounted increasingly on 1956 models.

The Tire and Rim Industry-approved TR500 series of Clamp-in Valves are now being manufactured in volume and are going on a large number of truck rims. The clamp-in principle of attaching the valve to the rim does not alter the method of putting air in the tires. It continues to be the tire inflation principle developed by Schrader over 50 years ago—as described on the opposite page.

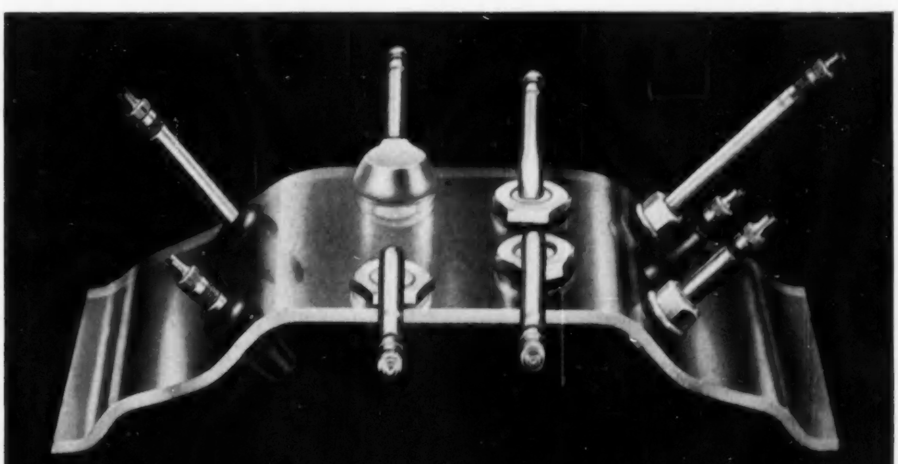
Eventually most vehicles will run on tubeless tires. The changeover will take place as soon as Industry rim-valve-tire requirements, now on test, are standardized for different vehicles. Because of Industry cooperation in the development, testing and production of *all three:* a better valve, better rim, better tire . . . motor vehicle transportation continues to improve!

As new problems are faced in tire valve development, you can be sure Schrader will continually be helping with research, design and production to provide quality valves that fit the job.

TR500 SERIES
CLAMP-IN 3-WAY SEAL



TYPICAL TUBELESS TRUCK VALVES IN ACTUAL USE OR ON TEST.



A. SCHRADER'S SON, *Division of Scovill Manufacturing Company, Incorporated*
470 Vanderbilt Avenue, Brooklyn 38, New York

FIRST NAME IN TIRE VALVES

FOR ORIGINAL EQUIPMENT AND REPLACEMENT

Packard's large staff of experienced engineers is at your service to help solve cable problems most efficiently. And when the problems are solved and your needs determined, Packard Electric—with facilities for the daily production of more than 7,000,000 feet of cable and 800,000 wiring assemblies—will assure you dependable, on-time delivery.

Consider Packard as a source

Packard cable and wiring harnesses meet or exceed applicable specifications. They have proved themselves time and again in meeting and standing up to the most exacting demands in automotive and aircraft service. Packard Electric's experience, skill and productive ability often result in substantial savings to customers. Possibly the savings to you could be substantial too.

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Packard

Packard Electric Division, General Motors, Warren, Ohio
Offices in Detroit, Chicago, and Oakland, California

AVIATION, AUTOMOTIVE AND APPLIANCE WIRING

WHY NOT

Reinforced Vibrin for...

...complete door liners? ...rigid automobile tops? ...station wagon trim?

Easily formed door liners of reinforced Vibrin polyester would even include the arm rest. Practically mar-proof, they need no painting or upholstering. And they're always pleasant to the touch. **Translucent tops** of reinforced Vibrin provide pleasant, uniformly lit interiors without uncomfortable heat ...without sun glare. They can be permanently mounted or removable. **Wood-like grained station wagon trim** of reinforced Vibrin can't rot...never needs waxing or other protection. Formed to stylish contours it serves as additional protection for the sides.

...seat back panels? ...hood and trunk sections? ...grille work for trucks?

Attractive, scuff-proof seat back panels of Vibrin need no upholstery—wipe clean with a cloth—give all the protection desired. **Hood sections** of reinforced Vibrin would help eliminate engine noise as well as problems in counterbalancing, in hinging, in fabricating complex fairings for air intakes.

Why not reinforced Vibrin for truck fenders, cab-over-engines? Why not reinforced Vibrin grille work? Why not window trim, instrument panels, seat bases...?

Why not reinforced Vibrin® for the many applications where its strength, light weight, beauty, and formability can not only improve your product and add to its saleability, but reduce your costs at the same time!

Reinforced Vibrin is...

- dent-proof
- rust-proof
- rot-proof
- stronger than steel by weight
- resistant to gas, oil, most chemicals
- highly resistant to abrasion and impact
- unharmed by extremes of weather
- extremely light in weight
- integrally colored
- easily molded to complex contours

Write for our free booklets on the molding, use, and many advantages of reinforced Vibrin polyester.

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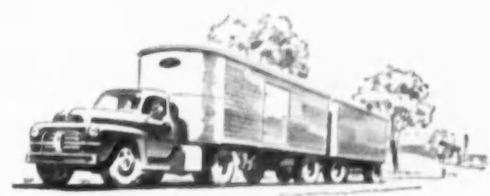
REDUCED MAINTENANCE • REDUCED STRESS AND WEAR ON ENGINE AND TRUCK PARTS

EATON 2-SPEED AXLES CUT COSTS FOR TRUCK OPERATORS

SAFER OPERATION • LESS TIME IN THE SHOP • HIGHER TRADE-IN VALUE • MORE PAYLOAD

The experience of thousands of truck operators in practically every industry shows that trucks equipped with Eaton 2-Speeds haul more, quicker, at lower cost per mile; spend more hours on the job, less time in the shop; last longer, and are worth more on the trade-in. Performance records prove that with reduced operating and maintenance costs, Eaton 2-Speeds pay for themselves over and over.



More than Two Million Eaton Axles in Trucks Today!

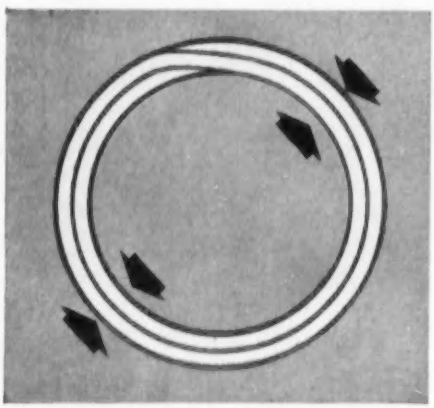
EATON AXLE DIVISION MANUFACTURING COMPANY CLEVELAND, OHIO

PRODUCTS: Sodium Cooled, Poppet, and Free Valves • Tappets • Hydraulic Valve Lifters • Valve Seat Inserts • Jet Engine Parts • Rotor Pumps • Motor Truck Axles • Permanent Mold Gray Iron Castings • Heater-Defroster Units • Snap Rings Springtites • Spring Washers • Cold Drawn Steel • Stampings • Leaf and Coil Springs • Dynamatic Drives, Brakes, Dynamometers

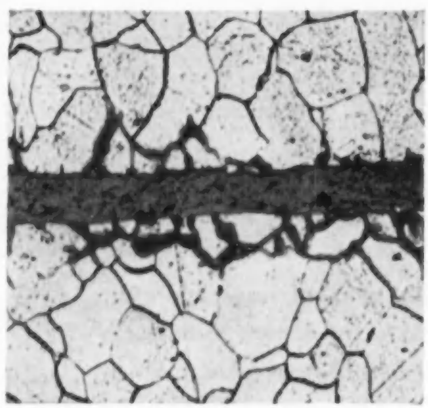
Only Bundyweld steel tubing

Here's why Bundyweld STEEL Tubing is used on 95% of today's cars

The illustrations below reveal why Bundyweld is specified by automotive manufacturers where strength and durability of tubing are essential. Bundyweld is the only tubing double-walled from a single metal strip. This exclusive process gives Bundyweld superior strength properties. Yet, because of the conditions under which Bundyweld is brazed and cooled, it is uniform and easy to fabricate.

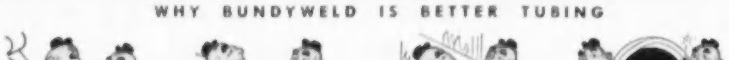


With Bundyweld's beveled edges and single close-tolerance strip, there's no inside bead. The tubing is uniformly smooth, both inside and out. It fabricates easily; can be bent to short radii. Copper coating, inside and out, facilitates soldering and brazing operations.



This view of Bundyweld's copper bond (enlarged 300 times) shows how the copper actually alloys with the steel . . . through 360° of wall contact. That's the secret of Bundyweld's outstanding resistance to high pressure and vibration fatigue.

WHY BUNDYWELD IS BETTER TUBING



Bundyweld starts as a single strip of copper-coated steel. Then it's . . .

continuously rolled twice around laterally into a tube of uniform thickness, and

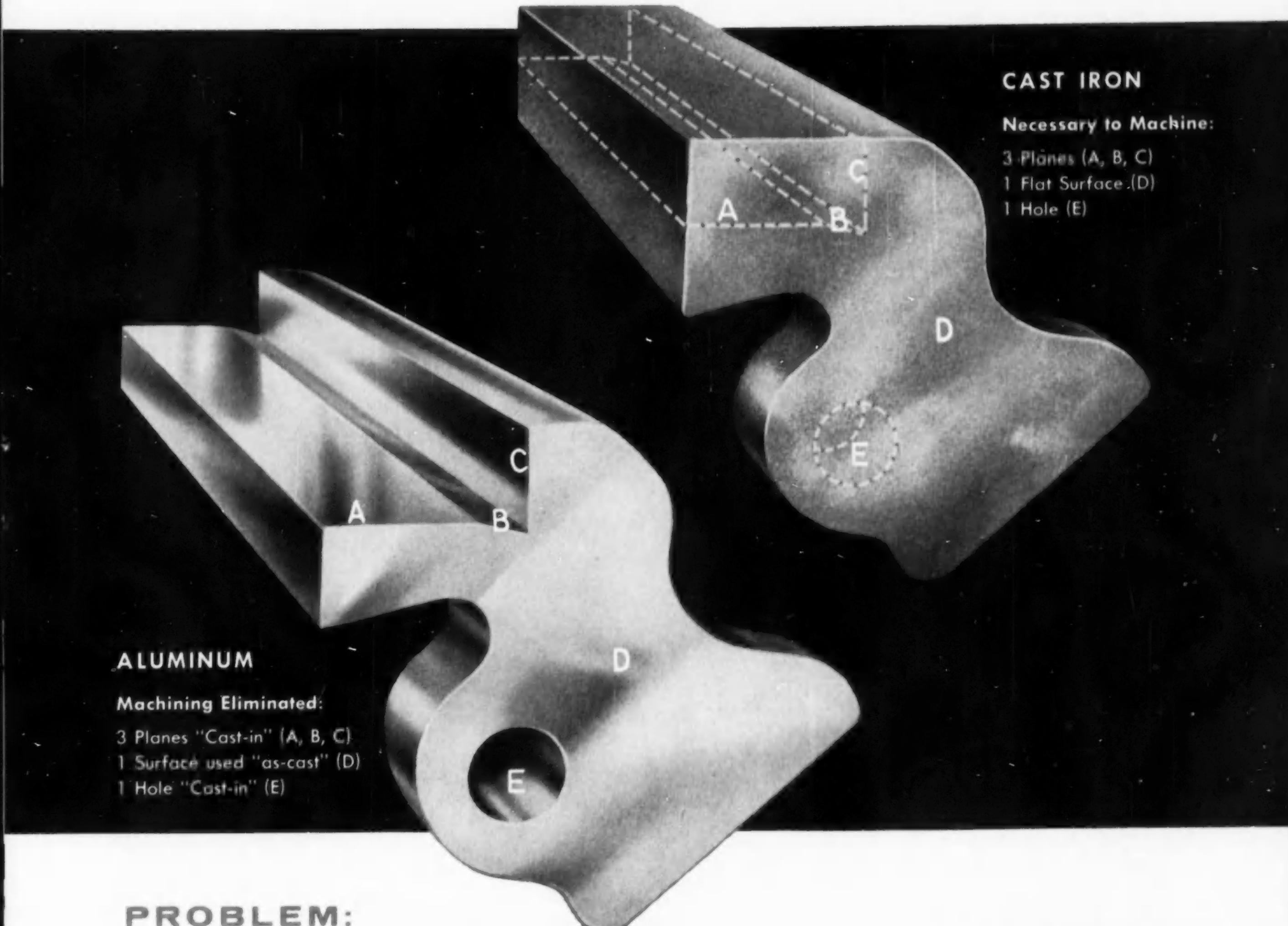
passed through a furnace. Copper coating fuses with steel. Result . . .

Bundyweld, double-walled and brazed through 360° of wall contact.

←

NOTE the exclusive Bundy-developed beveled edges, which afford a smoother joint, absence of bead, and less chance for any leakage.

The Case of the Costly Casting



PROBLEM:

A manufacturer recently found the cost of machining a certain cast iron part had become prohibitive. How could costs be reduced?

SOLUTION:

Investigation showed aluminum permanent mold castings of the same shape would reduce machining time about 10%. But Bohn engineers went several steps further. Working closely with the manufacturer, they re-engineered the job and re-designed the cast shape. This eliminated five machining operations.

RESULT:

Machining time was reduced 80%.

BOHN

ALUMINUM & BRASS CORPORATION

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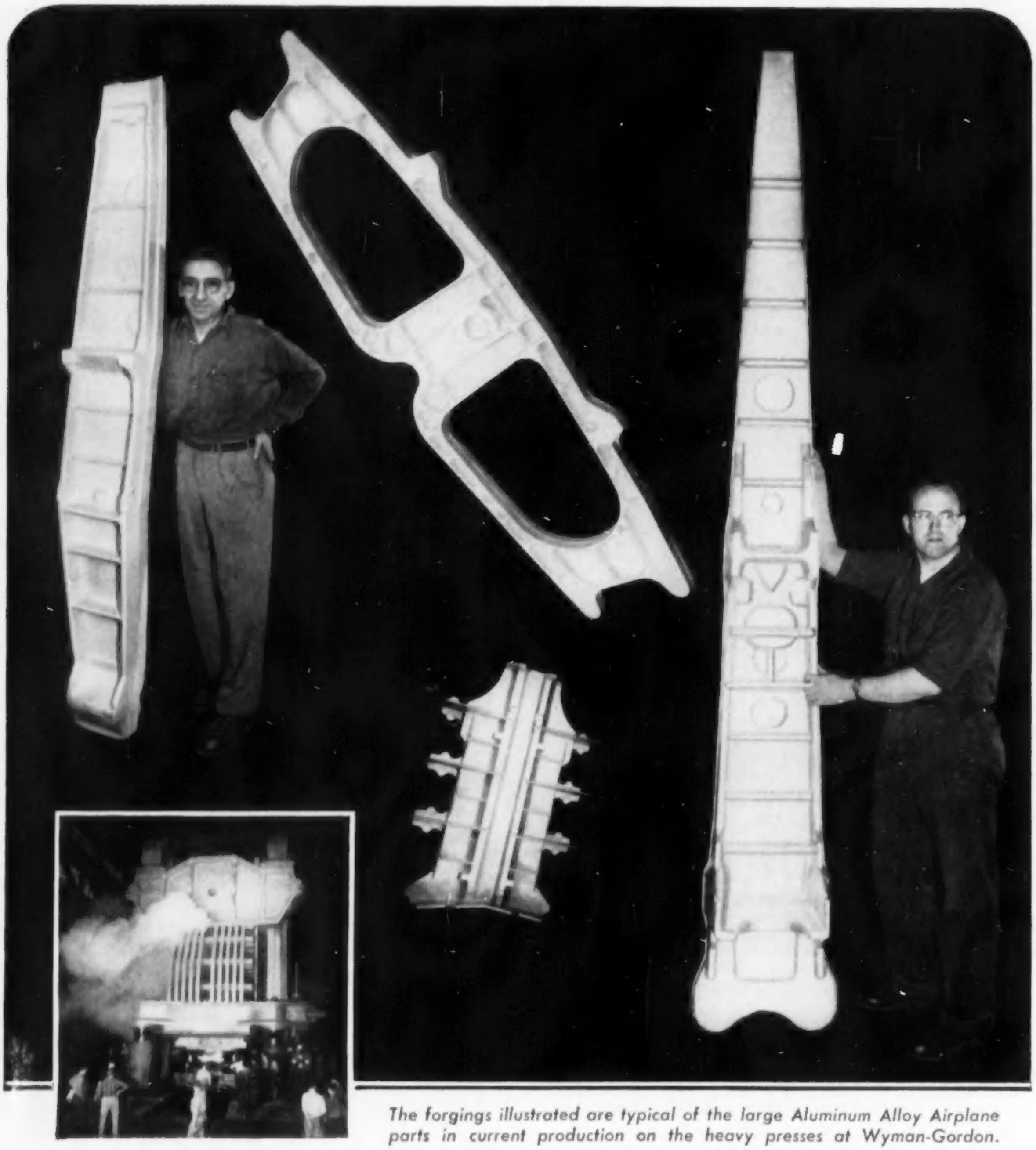

The forgings illustrated are typical of the large Aluminum Alloy Airplane parts in current production on the heavy presses at Wyman-Gordon.

Milwaukee, Minneapolis, Moline, New York, Philadelphia, Rochester, St. Louis

How Holley's rolling laboratories test the fuel metering devices for tomorrow's cars and trucks

When a carburetor prototype or an important design change has proven its merit at Holley's "inside" research lab, it's sent to the world's toughest proving ground—the streets and highways of America.

Holley's inside research and development laboratory—one of the best equipped in the automotive industry—can tell only part of the story. Cold room, dynamometer, and air box testing simulate to a high degree the driving conditions a carburetor prototype will ultimately face. Even before these physical tests, an electronic computer—Holley acquired one of the first in the automotive industry—checks and tabulates design theory from drawing board specifications.

But ultimately, the most conclusive test is found on city streets, country roads and super highways.

Holley engineers frequently average 500 miles a day for ten days measuring a carburetor's performance in heavy traffic or on super highways.

The results of this concentration on research both "inside" and "outside" have helped to bring Holley equipped cars 5 sweepstakes champions in Mobilgas Economy Runs and 3 straight Heavy Stock Car winners in the fabulous Pan American Road Race.

HOLLEY Carburetor Co.

FOR MORE THAN HALF A CENTURY—ORIGINAL EQUIPMENT MANUFACTURERS FOR THE AUTOMOTIVE INDUSTRY.

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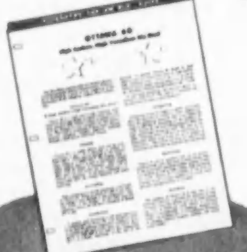
OTTAWA 60

for DRAW DIES is *NEW*... and *NEWS*!



Write for a copy of the OTTAWA 60 BLUE SHEET

This Blue Sheet contains certified data on the physical characteristics of Ottawa 60, prepared from carefully checked laboratory and field service tests. All the information you'll need on methods of handling and heat treatment, etc.

ADDRESS DEPT. SA-69

In fact, this exclusive Allegheny Ludlum-developed die steel is mighty *good* news for any user of draw dies. Ottawa 60 is a high-carbon, high-vanadium analysis, initially designed for the primary purpose of drawing stainless steel.

In that service, Ottawa 60 does just exactly what it was developed to do: it performs without galling or pickup, and shows exceptional wear-resistance—as a long list of successful applications will prove to you. Two of them are illustrated above: a stainless hinge and a stainless sundae server.

But Ottawa 60 is a top performer on any draw die application! Also illustrated above are the two draws on a transformer housing of .037" gauge SAE 1010 strip. After more than 25,000 pieces—*over 12 times any previous runs*—there was still no sign of pickup, or of wear on the Ottawa 60 punch or die.

This analysis can solve your draw-die problem jobs—or reduce your costs on almost any drawing operation. ● Call on our Mill Service Staff for any assistance. ***Allegheny Ludlum Steel Corporation, Oliver Bldg., Pittsburgh 22, Pa.***

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Allegheny Ludlum

WSD 5581

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AERONAUTICAL INFORMATION REPORTS ON AIRCRAFT GREASES (DATED MARCH 1, 1955)

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Having trouble with O-ring failures? Nibbled, worn, battered, leaking like a sieve? Whether your problem is in the basic design or compound, it's time to call a Parker representative.

Parker field engineers are trained trouble shooters. There is one in your area. From Parker you can get exactly the *right* O-ring and gland design for your *specific* application.

Compare Parker O-rings with any other make. You'll find that Parker O-rings are precision-molded of superior compounds. They have been developed as the result of years of experimentation to provide the proper elongation, tensile strength, compression set ratings, resistance to oils, fuels, chemicals, and high and low temperatures. Laboratory and service tests make sure that all rated characteristics are held.

Call your Parker representative for assistance. Mail the coupon for *free* technical bulletins about O-rings.

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The Parker Appliance Company
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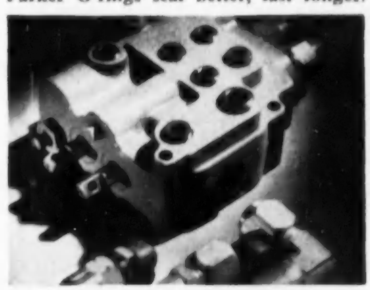
Heat failure was caused by a combination of compound and design problems. Parker field engineer was called. Leaks were stopped . . . and complaints ended.

Compare actual samples. Ask your Parker representative to check your specifications and design. Let him prove how Parker O-rings seal better, last longer.



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- ☐ Materials for O-rings Bul. 5705 B1
- ☐ O-ring Operation Factors Bul. 5705 B2
- ☐ Thermal Properties Bul. 5705 B4
- ☐ Rotary Seal for High Speed Bul. 5705 B5
- ☐ Aircraft Hydraulic Packings Bul. 5712 B1

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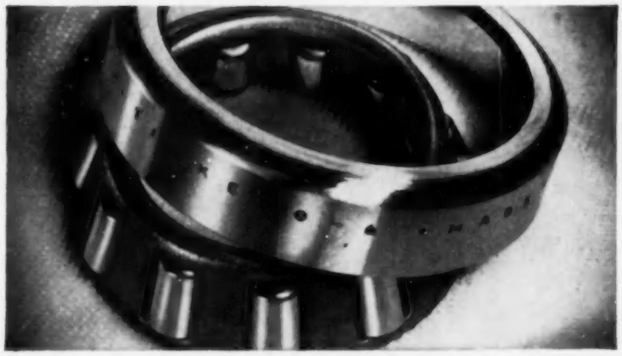

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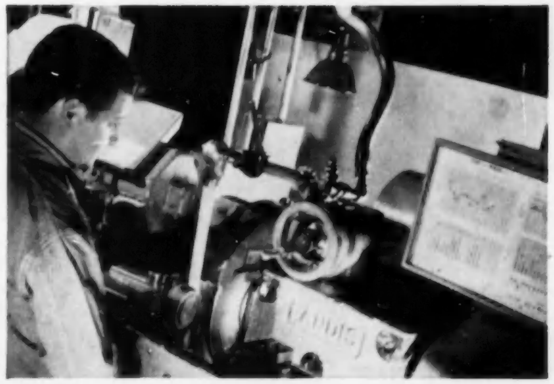
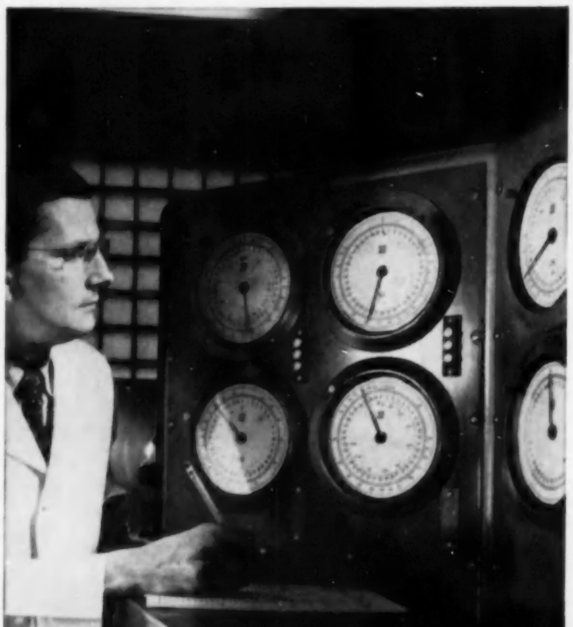


CATCHING REJECTS before they happen is the job of our statistical quality control. For instance, a chart like this on every grinder plots the dimensions of several cups or cones from each lot. Trends towards off-size bearing parts show up instantly. This is another step we take to make Timken bearings the No. 1 value in your cars' moving parts—the vital zone.



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